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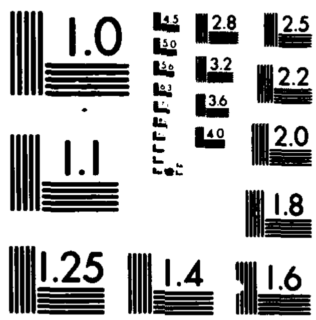
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ENERGY SAVING DEVICES FOR GAS FURNACES

FINAL REPORT

REPORT FESA-TS-2072

ENERGY SAVING DEVICES FOR GAS FURNACES

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Three devices designed for attachment to gas furnace flues were tested under specified conditions to measure their effect on the thermal efficiency of one furnace. An automatic vent damper (AVD) had no appreciable effect on furnace thermal efficiency. The tests confirmed the claims for this device, that it is of benefit only in reducing air exfiltration while the furnace is off. A vent restrictor and an extended draft hood were of varying effectiveness under the test conditions employed. The performance of these two devices showed that additional testing including furnace derating would be required to properly → next page

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evaluate and compare these devices. Any consideration of retrofit vent devices should be reviewed with building officials, furnace manufacturer, device manufacturer and local utility with regard to building codes, mechanical codes, furnace warranty, and safety factors.

SUMMARY

Three different vent connector modification devices for natural gas fueled furnaces were evaluated to measure their effect on the thermal efficiency of a standard 110,000 BTU furnace, using the specific testing regimen described in Appendix A. This work was conducted at Colorado State University. An automatic vent damper (AVD) had no major effect on thermal efficiency of the furnace. A vent restrictor had no effect on thermal efficiency and a retrofit draft hood called Thrifty-Vent had only a small, varying effect on thermal efficiency, under the conditions of the testing procedures used.

The data obtained during testing clearly showed that a furnace vent device should not be judged solely on the basis of its effect on thermal efficiency of a furnace. The data also suggested that some vent devices may perform best when combined with derating of the furnace. Further, it appeared that the final judgement of the benefit of a particular device can best be made in a "real house" situation. This may be impractical but, nevertheless the furnace-house interaction with regard to, for example, house stack effect during off cycle exfiltration may have a strong bearing on the real benefit of some devices.

An example of an annual gas cost savings was calculated for each device combining the change in furnace efficiency with the benefit/penalty of changes in room air exfiltration during both on and off cycles. This led to the conclusion that the AVD may be cost effective when installed in high wind areas. The savings should increase as vent diameter increases over 5 inches and as furnace off time increases. It was reported that the particular device tested, a Penn Q-16, is recommended for retrofit use only when combined with an intermittent ignition device. This combination is likely to be cost effective in any high wind area.

A simple vent restrictor showed major cost savings only in the condition of high wind area combined with the furnace located in unheated space. The vent restrictor increased exhaust temperature at the furnace outlet which may indicate a higher temperature in the furnace heat exchanger. This suggested the possibility of reduced heat exchanger life, due

to the risk of increased fatigue. On the positive side, the higher temperatures suggested that a vent restrictor might prove to be a valuable device when combined with a furnace derating.

The Thrifty-Vent was the most effective of the three devices, when thermal efficiency and air flow were combined. Its cost effectiveness was not explored as installed cost information was not obtained. Like the vent restrictor, Thrifty-Vent was most effective when the furnace was located in unheated space. It also improved overall savings as furnace duty cycle increased. This latter fact combined with much higher exhaust temperatures and reduced vent temperature suggested that Thrifty-Vent will perform best when combined with a furnace derating.

It is recommended that the tests including baseline, vent restrictor and Thrifty-Vent be repeated at several levels of furnace derating. Only in this way can the real benefits of Thrifty-Vent and a vent restrictor be determined.

Caution

Installation of vent devices on gas furnaces may alter the operating characteristics of the furnaces. Changes in the temperatures of flue and vent gases as well as in the heat exchanger may occur. The safe installation and operation of any vent device should be carefully and fully reviewed with local building officials, the gas utility and a technical representative of the furnace manufacturer. Two potential hazards which were suggested by temperature measurements made during this study were (1) increased risk of combustion gases condensing in the vent which could accelerate corrosion of the vent or even the furnace itself, and (2) an increased heat exchanger temperature could accelerate "burn out" of the exchanger.

Safety considerations were not intended to be addressed by this study, thus there may be other potential hazards not disclosed by the test data.

PREFACE

The laboratory study of gas furnace stack devices was conducted at Colorado State University by Mr. Thomas E. Brisbane under the direction of Dr. S. Karaki, Professor of Civil Engineering and Director, Solar Energy Application Laboratory.

Mr. Harlan Bongard of Gas Masters of Colorado, Inc. volunteered to install the Thrifty-Vent.

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INTRODUCTION

The Final Report on Task Order No. 1 of Contract DAAK 70-78-D-0002 was a State of the Art Survey of vent and stack devices for combustion furnaces. This report provided guidelines for judging the desirability and economic benefit of installing certain of these devices on existing furnaces. Following this report, a desire was expressed to test, on a laboratory basis, three different vent devices for natural gas furnaces.

The objective under this Task Order 7 was to determine whether significant or practical increases in gas furnace thermal efficiencies measured on a cyclic basis could be obtained through the use of add-on vent devices. Some of these devices are being widely marketed and advertised in light of the steadily rising cost of energy, and it is of interest to have quantitative and qualitative knowledge as to their actual effectiveness.

The three devices tested are readily available and immediately applicable for use on a natural gas furnace with only minor modifications to the combustion products vent. Local codes may have a bearing on their use or installation. These devices include an extended draft hood, an automatic vent damper and a vent restrictor. All tested devices were to be compared with a "baseline", the basic furnace vented with no alterations, just as it would be set up and installed in typical homes.

Tests of furnace thermal efficiency were made for selected operating cycle times and temperatures in the vent and air ducts were measured at the various operating conditions. Both a heated furnace room and an unheated "crawl space" or furnace room were simulated in the tests. Also wind conditions which create higher drafts up the vent were simulated using a two speed fan. Each device was tested for the full array of comparable operating conditions and furnace thermal efficiencies for the various tests are compared.

This report describes comparisons of test results with the devices tested. Hypothetical energy cost savings are calculated from these results using, as an example, an assumption of heating hours, average ambient temperature and fuel cost. The test facility, test procedures and instrumentation are described in Appendix A. Key data for all tests are shown in Appendix B.

DISCUSSION

Testing Method

The test facility consisted of a "furnace room" 6 feet wide and 12 feet long and an instrumentation room the same size located immediately adjacent to the furnace room. The "room air" for the test furnace was supplied from laboratory space and heated air was delivered to the lab space at 8 feet above the floor and directed in an opposite direction from the cold air intake. The air temperature near the floor of the laboratory varied between 18° to 20°C which is representative of return air temperature in a heated building.

A regulated natural gas supply line already existed within the building and was easily extended to the test rooms for operation of the furnace. The combustion vent was run through an office balcony located immediately above the test rooms and vented into a space near the "roof" inside the building. A vent fan, operating whenever the furnace was operating, then pulled the combustion gases outside the lab building. To simulate a wind induced draft through the vent, a two speed vent fan was used. Combustion air for the furnace room was drawn in through a wall vent from the laboratory floor, well away from the "room air return" intake. For an unheated furnace room a package room air conditioner was installed in the furnace room to cool the furnace room down to about 10°C.

The furnace was a standard model which included a four speed blower and a standing gas pilot light and all burner controls, fan switches and 24 volt control wiring.

Each air duct monitor consisted of total and static pressure probe arrays which were separately manifolded to an outlet port.

The following data were taken for the analysis of each of the devices attached to the vent:

1. Temperatures through the system.

2. Air flow rates in the room air circuit.

3. Vent flow rates during and before burner operation.

4. Wet and dry bulb temperature.

5. Gas consumption during each run.

6. Electric power to the blower motor.

Data were recorded on magnetic digital tape.

A detailed description of each piece of testing equipment; the testing set-up; and data acquisition appears as Appendix A.

Testing

The testing matrix for each device was as follows:

Device under Test	Furnace Room	Vent Fan	Burner on % Duty Cycle
Baseline	Heated	Off	10, 20, 30, 50, 75, 100
Thrifty-Vent, Vent Restrictor, AVD		Low	10, 20, 30, 50, 75, 100
		High	10, 20, 30, 50, 75, 100
	Unheated	Off	10, 20, 30, 50, 75, 100
		Low	10, 20, 30, 50, 75, 100
		High	10, 20, 30, 50, 75, 100

The low and high fan settings produced very similar equivalent wind velocities ranging from about 40 MPH at the low setting to nearly 50 MPH at the high setting. The equivalent wind velocity ranges obtained at the low and high fan settings overlapped each other. The fan "off" setting was equivalent to a still air condition.

The magnetic digital tapes created during each day's run were turned into the University Computer Center which has a remote terminal and batch deck reader located at the Engineering Research Center. Once a tape was in the computer center, a simple and quick program was run to verify that the data on the tape were correct in format and that the tape was readable by the computer. At this point the data on the tape were transferred to a "master" tape contained internally at the computer center. All actual data anlaysis was done from the master tape.

All data channels were sampled at a rate of about one scan per 1.5 seconds resulting in about 1200 discreet points per data channel for each 30 minute test run. The advantage of computer analysis in this case is obvious in the quanitity of data processed during analysis. The short time interval between data samples thus permitted close tracking of rapid temperature changes in various parts of the system.

Calculations required knowledge of mass flow rates, temperatures and temperature differences. Thermocouple and thermopile outputs were recorded in elecromotive force and converted to temperatures and temperature differences. Voltage to temperature conversion was carried out using National Bureau of Standards[1] equations for fourth order fit of type T thermocouple response to temperatures in the range between 0°C and 400°C. The conversion equation can be expressed as follows:

$$T(^{\circ}C) = (2.5561297 \times 10^{-2})E - (6.1954869 \times 10^{-7})E^2 + (2.2181644 \times 10^{-11})E^3 - (3.5500900 \times 10^{-16})E^4$$

where T is temperature in °C
E is emf of thermocouple in microvolts

The program used the meter readings of gas consumed and barometric pressure and temperature to convert the gas quality to a total Btu input to the furnace. Thermal efficiency in percent was expressed as

$$\text{Thermal Efficiency} = 100 \times \frac{\text{Heat Delivered to Room System}}{\text{Heat Content of Gas Input}}$$

[1] NBS Monograph 125 (Thermocouple Reference Tables Based on the IPTS-68), U.S. Department of Commerce, National Bureau of Standards, March 1974.

This was determined using temperature differences in three different ways to provide checks on the calculations;

(1) Using the hot air delivery flow monitor and the room air system thermopile,

(2) using the cold return air monitor and the room air thermopile and

(3) using the hot air delivery flow monitor and the temperature in the circulating air ducts.

The first method was judged to be the most reliable of the three. The thermopile provided the better temperature differences because it averaged temperatures at nine points in the cross sections of the duct. The hot air monitor included air leakage into the circuit at the furnace and blower but the air is heated up by the heat exchanger. The latter two calculations provided checks on the first in the event of "bad" data in isolated channels or if gross errors occured in calculation.

The output from the computer program was a complete table of temperatures throughout the system, the heat flow rates calculated during blower operation, gas input Btu, Btu obtained from room air system, and the thermal efficiency of the system. A separate table was printed for every test condition. Appendix B summarizes these tables and includes data taken manually at the time of the test.

Results

Test runs began on February 7, 1979 and proceeded through March 26, 1979 with data taken as described in Appendix A for the baseline and three energy vent devices. Furnace thermal efficiencies using the devices as well as the baseline tests are presented in Table 1. Efficiencies were calculated using the room air delivery flow rates and the room air circuit thermopile as the basis for calculating the quantity of heat delivered from the furnace.

By comparing efficiencies in each column of Table 1, it appears that none of the energy saving devices had a major effect under the specific conditions of these tests. The "Thrifty Vent" (extended draft hood) did provide the greatest improvement in performance. Average thermal efficiency increased from an overall baseline value of 72.5 percent to an overall efficiency of 76.8 percent with the Thrift-Vent installed.

Point by point the improvements provided by each device compared to baseline efficiencies are listed in Table 2. Overall improvement offered by Thrifty-Vent was 4.3 percent. The Thrifty-Vent appeared to offer more improvment during the simulated wind conditions (fan on low or high). The improvement when no fan was used was 3.6 percent over baseline while the improvement in efficiency over baseline is 4.7 percent for the vent wind condition.

The induced wind effect on furnace efficiency was small and variable. The data suggest that furnace efficiency decreased slightly at higher wind velocities but the differences noted were very small for the unmodified furnace. Wind velocity interaction with vent devices is illustrated in Tables 5, 6, 7, and 8.

TABLE 1. FURNACE THERMAL EFFICIENCIES FOR BASELINE AND TEST DEVICES. EFFICIENCIES CALCULATED USING WARM AIR DELIVERY FLOW RATE AND DELTA TEMPERATURE MEASURED BY THERMOPILE.

Device	Furnace room temp.	Stack fan	Efficiency % (based on hot air monitor) Duty cycle 10%	20%	30%	50%	75%	100%
Baseline	Heated	Off	63.4	72.2	76.6	82.4	77.0	77.1
		Low	69.5	72.3	74.8	76.5	76.2	76.0
		High	66.5	71.2	75.3	75.6	76.8	80.2
72.5% overall	Unheated	Off	60.2	70.1	70.6	74.5	77.0	77.9
		Low	59.3	69.5	71.7	73.9	75.5	77.6
		High	54.5	65.9	69.2	72.1	74.2	75.0
Thrifty Vent	Heated	Off	66.6	76.6	80.3	81.3	82.0	80.5
		Low	66.4	78.8	80.3	80.7	81.9	81.8
		High	63.8	76.4	77.7	79.4	78.6	81.6
76.8% overall	Unheated	Off	60.1	73.1	77.2	80.2	81.6	82.9
		Low	61.1	73.4	76.3	79.7	81.7	82.9
		High	63.6	73.2	76.1	80.5	81.6	84.1
Automatic vent damper	Heated	Off	62.2	74.2	75.7	81.9	78.0	80.7
		Low	55.1	69.1	72.5	75.7	76.7	79.6
		High	61.5	74.6	78.0	79.6	78.9	79.5
73.4% overall	Unheated	Off	61.1	74.2	76.9	78.4	77.3	80.2
		Low	58.1	70.8	74.4	75.6	76.1	78.4
		High	59.6	68.5	70.3	73.9	76.7	77.2
Vent Restrictor	Heated	Off	65.9	71.7	75.8	76.1	77.2	79.0
		Low	62.6	74.0	72.0	74.5	77.6	77.6
		High	59.0	73.6	71.2	74.9	75.5	75.8
72.2% overall	Unheated	Off	61.8	69.1	75.5	74.1	76.3	77.6
		Low	60.0	67.7	72.2	73.9	80.0	78.0
		High	59.0	65.0	68.8	74.5	76.5	75.1

TABLE 2. IMPROVEMENTS IN FURNACE EFFICIENCY WITH THE TESTED DEVICES COMPARED TO BASELINE EFFICIENCIES.

Device	Furnace room temp.	Stack fan	Improvement in efficiency over baseline (%) — Duty cycle 10%	20%	30%	50%	75%	100%
Thrifty Vent	Heated	Off	3.2	4.4	3.7	-1.1	5.0	3.4
		Low	-3.1	6.5	5.5	4.2	5.7	5.8
		High	-2.7	5.2	2.4	3.8	1.8	1.4
4.3% overall	Unheated	Off	-0.1	3.0	6.6	5.7	4.6	5.0
		Low	1.8	3.9	4.6	5.8	6.2	5.3
		High	9.1	7.3	6.9	8.4	7.4	9.1
Automatic vent damper	Heated	Off	-1.2	2.0	-0.9	-0.5	1.0	3.6
		Low	---	-3.2	-2.3	-0.8	0.5	3.6
		High	-5.0	3.4	2.7	4.0	2.1	-0.7
1.4% overall	Unheated	Off	0.9	4.1	6.3	3.9	0.3	2.3
		Low	-1.2	1.3	2.7	1.7	0.6	0.8
		High	5.1	2.6	1.1	1.8	2.5	2.2
Vent restrictor	Heated	Off	2.5	-0.5	-0.8	-6.3	0.2	1.9
		Low	-6.9	1.7	-2.8	-2.0	1.4	1.6
		High	-7.5	2.4	-4.1	-0.7	-1.3	-4.4
-0.3% overall	Unheated	Off	1.6	-1.0	4.9	-0.4	-0.7	-0.3
		Low	0.7	-1.8	0.5	0.0	4.5	0.4
		High	4.5	-0.9	-0.4	2.4	2.3	0.1

The automatic vent damper and the vent restrictor did not cause any marked improvement in furnace thermal efficiency and in fact the overall performance with the vent restrictor was very slightly less than the overall baseline efficiencies (0.3%).

An important factor responsible for some of the data scatter was in blower operation. Since the blower was operated by temperatures within the heat exchanger, activated by a thermal switch, the "turn on" and "turn off" points of blower did not remain consistent through various tests. The temperature ranges which the heat exchanger experienced appeared to vary from device to device and blower cycling varied as a result. A one minute difference in blower cycling time can result in a one percent difference in calculated thermal efficiency.

Table 3 presents the furnace efficiencies calculated using the cold return air flow monitor. Flow rates measured in the return air duct were slightly lower than measured in the warm air delivery duct due to air leakage at the furnace and blower cabinet. The net result was to reduce slightly the efficiencies obtained, but the trend of the data followed and served as a check on the previous results using the hot air flow rates.

TABLE 3. FURNACE EFFICIENCIES FOR BASELINE AND TEST DEVICES BASED ON COLD AIR RETURN FLOW RATES.

Device	Furnace room temp.	Stack fan	Efficiency (%) (based on cold air monitor) Duty cycle 10%	20%	30%	50%	75%	100%
Baseline	Heated	Off	59.2	67.9	72.2	77.7	71.4	71.8
		Low	64.9	67.2	69.4	72.1	70.8	70.9
		High	62.3	67.1	71.4	71.2	73.2	----
	Unheated	Off	56.4	66.4	67.0	69.2	73.0	74.7
		Low	53.2	61.6	65.1	66.1	67.6	70.0
		High	49.9	58.7	61.9	66.1	66.7	67.5
Thrifty-Vent	Heated	Off	61.2	69.3	73.0	75.5	74.6	73.5
		Low	59.7	69.8	73.0	73.4	74.7	74.9
		High	57.2	69.4	70.5	72.3	71.6	76.5
	Unheated	Off	52.6	65.9	69.9	72.9	74.3	74.1
		Low	53.7	66.2	69.1	72.4	76.1	75.8
		High	57.2	67.7	70.6	73.5	76.2	76.4
Automatic vent damper	Heated	Off	57.7	67.1	70.0	76.0	70.8	72.0
		Low	52.5*	65.7	68.9	72.3	74.9	74.8
		High	57.7	70.4	73.9	75.8	75.1	74.3
	Unheated	Off	58.4	71.5	72.7	74.8	76.0	77.1
		Low	54.5	66.8	72.1	72.4	72.4	77.0
		High	55.8	64.5	67.8	71.6	73.0	73.7
Vent restrictor	Heated	Off	61.5	67.9	71.7	70.5	73.2	73.7
		Low	58.7	69.8	68.3	70.6	73.4	74.1
		High	55.5	69.5	67.7	71.3	72.0	72.5
	Unheated	Off	57.8	65.1	71.4	70.3	72.3	74.2
		Low	56.1	63.9	68.2	70.3	76.1	74.6
		High	55.2	61.5	67.1	70.8	71.2	70.8

Time based changes in air temperatures in room delivery duct and vent position just above flue were examined for selected runs at 30 percent duty cycle. Figure 1 is a graph of temperature versus time for the baseline compared with the Thrifty-Vent. As an indication of heat exchanger temperature the temperature of the flue gases at the outlet of the furnace was plotted for each case. Also, the temperature of the room air being delivered out of the furnace heat exchanger was plotted shown as a function of time.

The Thrifty-Vent caused a marked rise in temperature of the flue gases. For the 30 percent duty cycle in Figure 1, the temperature of the flue gases reaches nearly 178°C, where the corresponding temperature for baseline is only 130°C or a difference of 48°C between the two cases. The lower graphs representing temperature of room air delivered from the furnace are virtually identical, however. The increased temperature of the flue gases was not reflected by a proportionally higher temperature of air flow to the room delivery system.

Figure 2 is a graph depicting the same test condition as Figure 1 but this time comparing the automatic vent damper with the baseline. As expected, there were few differences between the baseline and the AVD in terms of temperatures at the flue gas outlet from the furnace or room air delivery while the burner was on. The major difference occured after the thermostat initiated a command to shut the burner off. With the AVD, a shift in the position of the damper blade also initiates burner shut off.

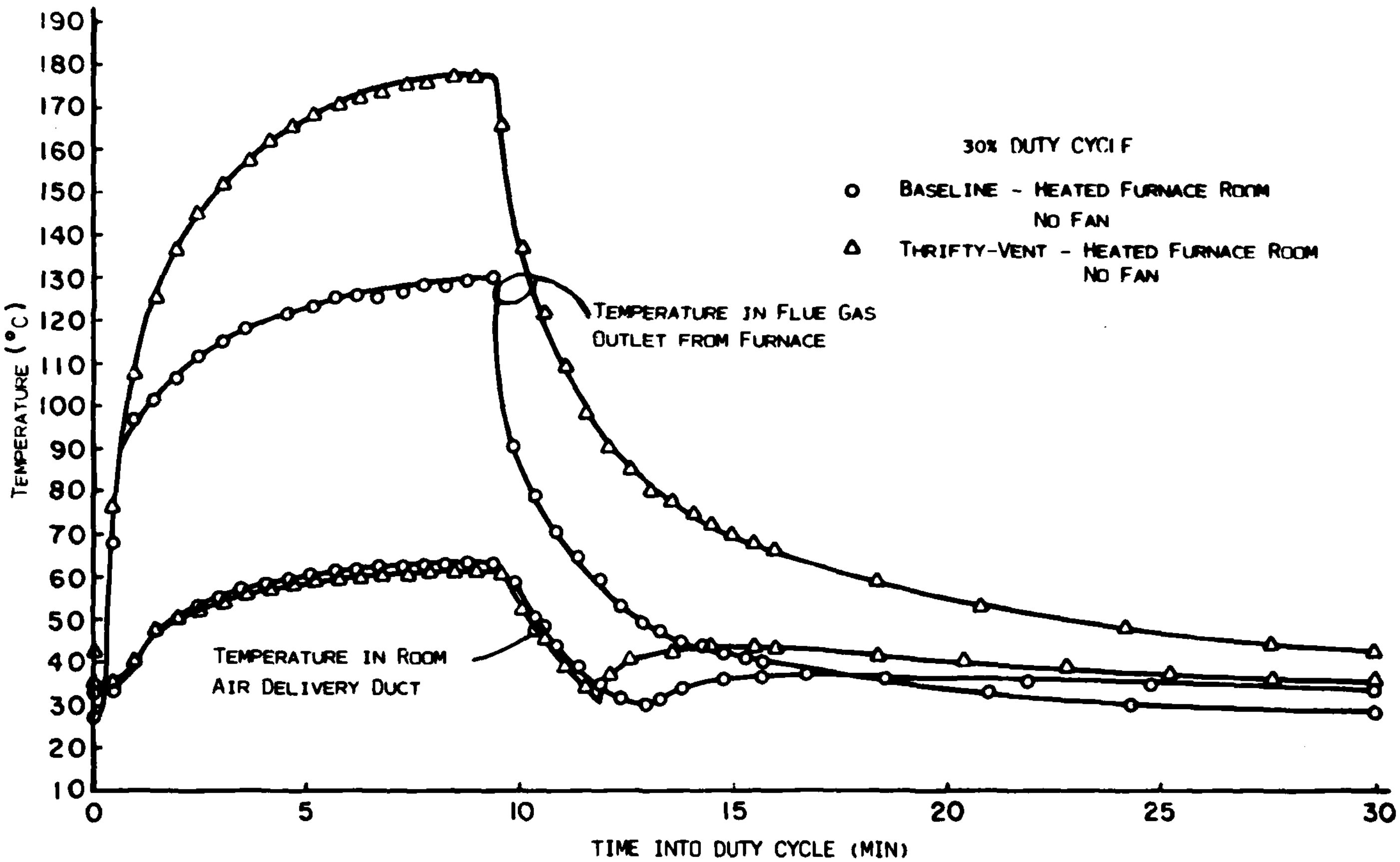


FIGURE 1. PLOT OF TEMPERATURES COMPARING BASELINE WITH THRIFTY-VENT, FLUE GAS AND ROOM AIR DELIVERY TEMPERATURES DURING 30% DUTY CYCLE.

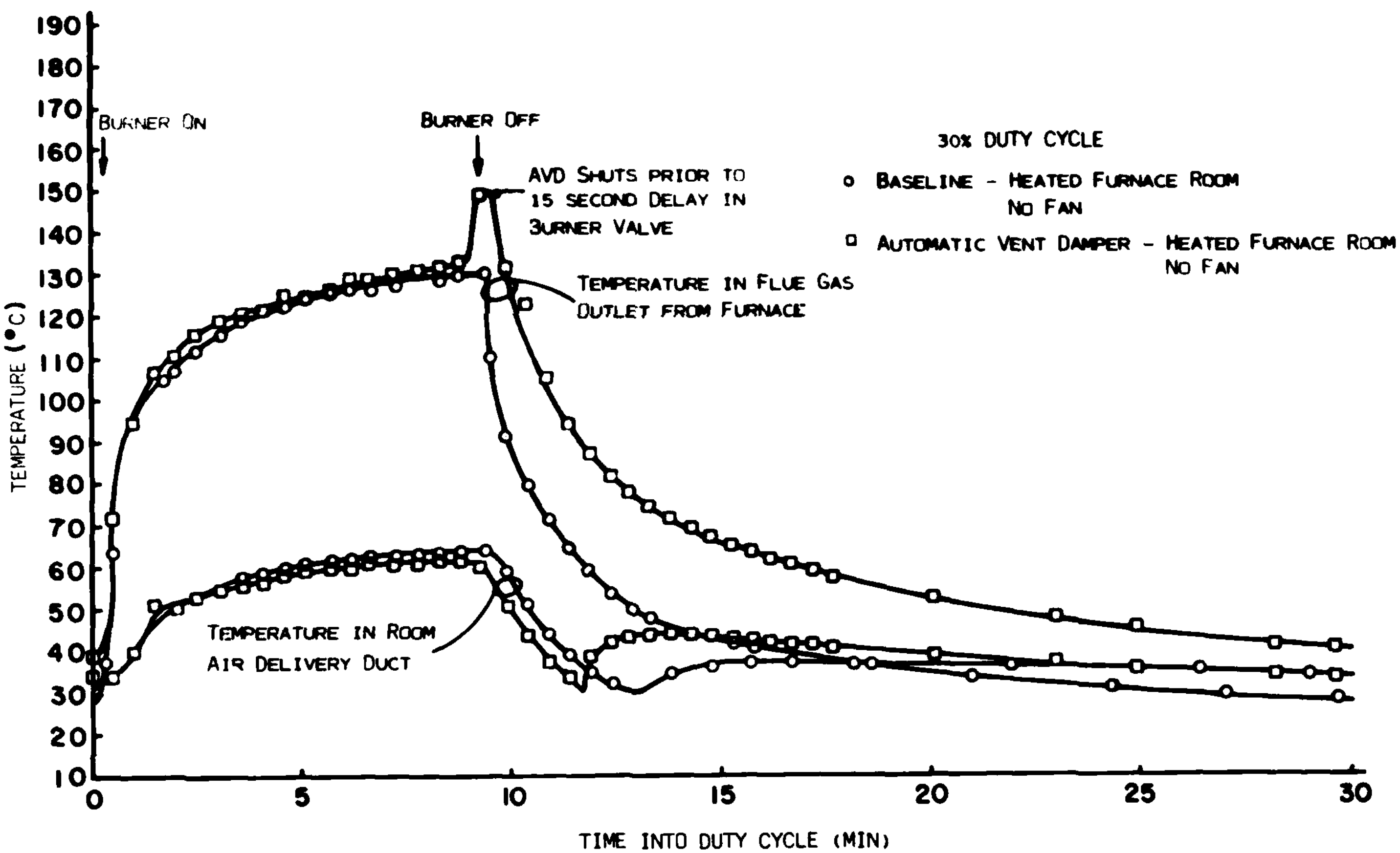


FIGURE 2. PLOT OF TEMPERATURES COMPARING BASELINE WITH AUTOMATIC VENT DAMPER, FLUE GAS AND ROOM AIR DELIVERY TEMPERATURES DURING 30% DUTY CYCLE.

The burner solenoid on the test furnace had a time delay of approximately 15 seconds when the furnace was operated "as installed". When the Q-16AVD was installed this time delay increased to approximately 25 seconds. This factor may have been influenced by installing the Q-16 AVD without an intermittent ignition device which is required for manufacturer's approved home furnace installation. Thus there is a short period of operation when the automatic vent damper was actually closing while the burner was still in operation. This caused the temperature spike seen on the graph of flue gas temperatures right at burner shut off (9.5 minutes into cycle). Further, once the AVD was in the closed position it served to hold heat in the heat exchanger as measured at thermocouple TC-3, Figure A-3 over a longer period of time. This fact did not alter the blower turn off point, however, as can be seen on the curve of temperatures in the room air delivery system. The room air delivery temperatures were not altered during furnace operation which was not surprising since the vent was essentially unaltered when the AVD was open.

The temperatures for the baseline and the 3-1/4-inch diameter opening vent restrictor are in Figure 3. Temperatures just above the heat exchanger are considerably higher than the baseline temperatures. At the end of the burner cycle the flue gas temperature at TC-3 was 35°C hotter than the baseline condition, or 130°C compared with the 165°C for the vent restrictor. The temperatures are not so high as those obtained with Thrifty-Vent.

As was the case for the Thrifty-Vent, the vent restrictor did not increase temperatures to the room air delivery system. The temperature curves for the baseline and vent restrictor are virtually identical resulting in no notable increase in heat delivered to the room air circuit.

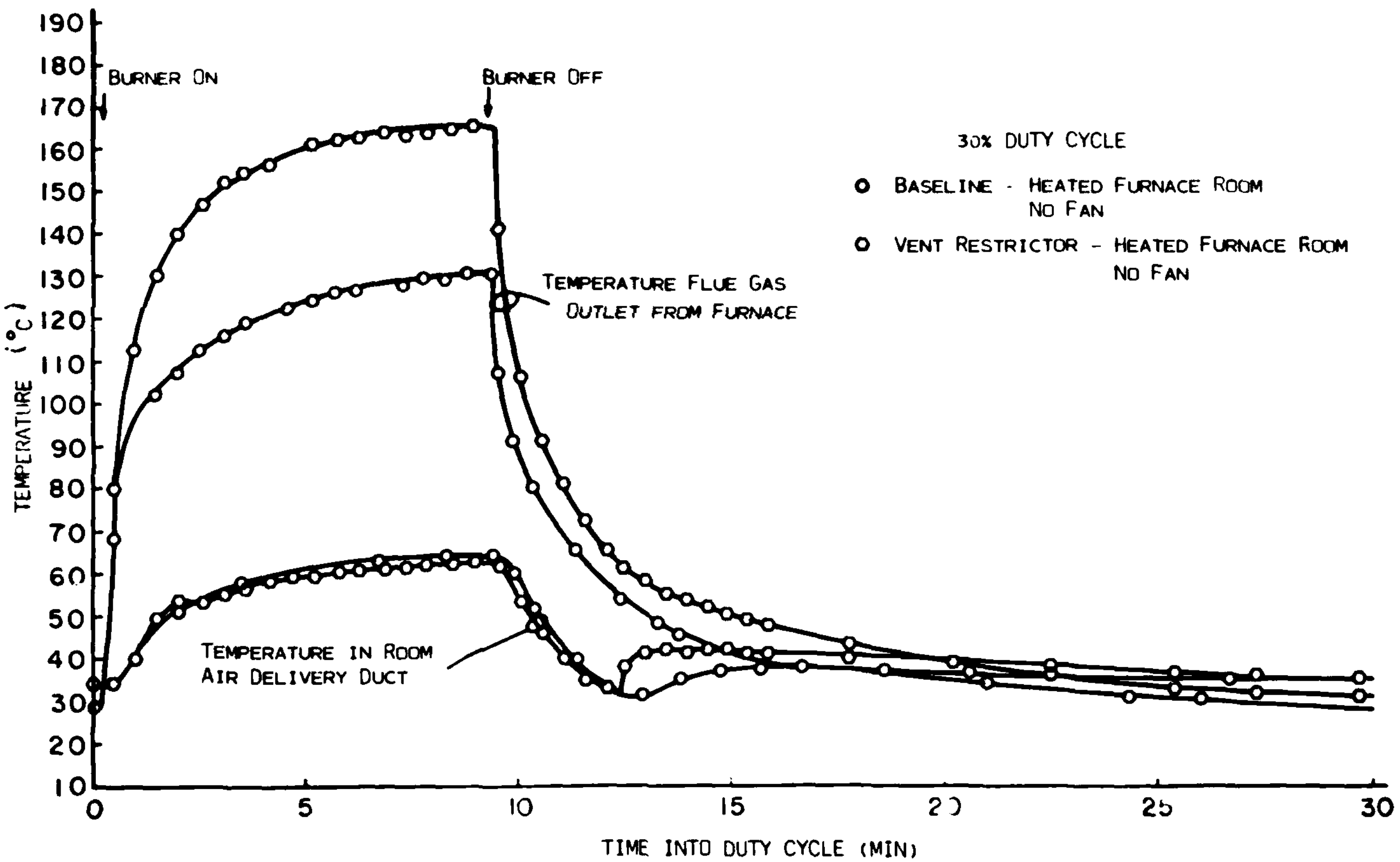


FIGURE 3. PLOT OF TEMPERATURES COMPARING BASELINE WITH VENT RESTRICTOR, FLUE GAS AND ROOM AIR DELIVERY TEMPERATURES DURING 30% DUTY CYCLE.

The vent temperatures resulting from the use of the devices are the most significant differences evidenced from the various tests. Two of the devices, the Thrifty-Vent and the automatic vent damper, tended to raise the vent temperatures markedly compared to baselines thereby possibly tending to hold heat in the heat exchanger for a longer period of time after burner shut-down. In neither case, however, did this alter the blower shut off point so that the heat in the exchanger was not delivered to the room air supply duct. Some gain might conceivably be realized by extending blower operation further after burner shut off. A timer delay might prove satisfactory for blower shut down.

In the test furnace, the blower "turn on" and blower "turn off" adjustments were located on the same scale making it physically impossible to adjust "turn off" much further down in temperature without also adjusting the "turn on" point downward. While this might help efficiencies of some of the devices, it would also be likely to improve the efficiency of the baseline. These blower operating points were assumed fixed within the constraints of the present tests.

The total temperature difference experienced by the heat exchanger for the cycle shown in the baseline was about 101°C while for the Thrifty-Vent and the Vent restrictor it was 135°C. Since life expectancy of the heat exchanger may be related to the stresses induced by temperature changes during cycling, both the Thrifty-Vent and the vent restrictor might contribute to problems in this repect, and under the conditions of these specific tests.

The temperatures of the flue gases on the atmospheric side of the devices are also of some interest. Too low a temperature at this point can cause water vapor to condense inside the vent prior to being exhausted. Ideally the dew point temperature of the flue gases should not be reached until these gases have passed out of the vent system into the open air. Figure 4 is a plot of these temperatures in the vent after the energy saving devices.

At this point in the vent system, the vent restrictor and the Thrifty-Vent plot on opposite sides of the baseline curve. (The AVD should have no effect on temperatures at this point in the vent except after burner shut off and is not plotted.) The Thrifty-Vent exhibited a 45°C lower temperature at the outlet to the vent at furnace shut off compared to the baseline. At the same point the vent restrictor results in a 20°C temperature increase.

To what extent the lower temperature of vent gas from the Thrifty-Vent might represent a water condensate potential would be a function of many things such as vent configuration, temperature of outside air, temperature of dilution air, etc. The vent restrictor with higher temperatures than baseline in the vent system reduces the potential for condensate. The ability of an existing vent system to safely handle the higher temperature gases might come into question, however. The temperatures reached under the conditions of this test were about 70°F under the maximum for type B vent systems.

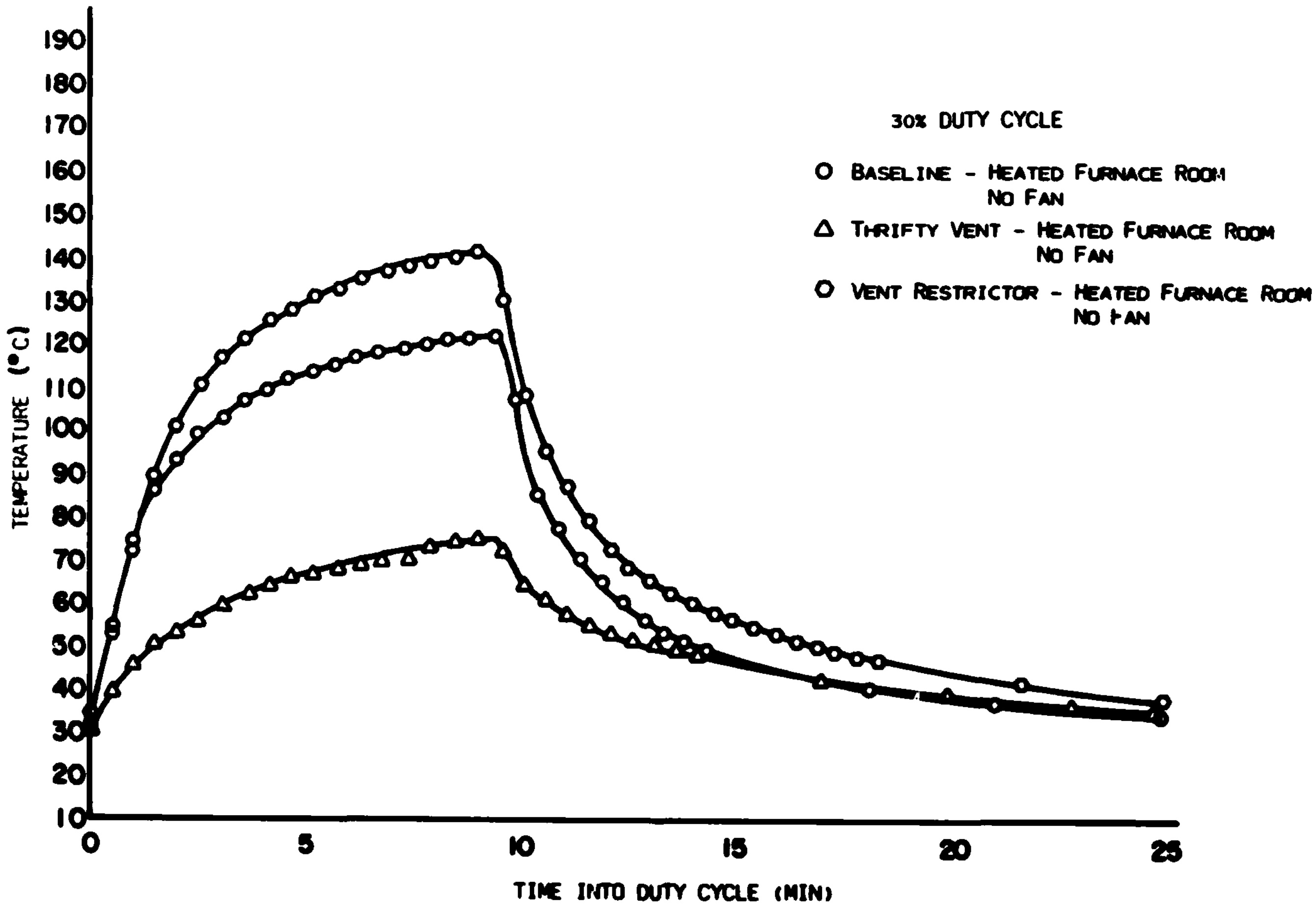


FIGURE 4. PLOT OF TEMPERATURES COMPARING BASELINE, VENT RESTRICTOR AND THRIFTY-VENT TEMPERATURES IN THE VENT AFTER THE DEVICES.

Because the Automatic Vent Damper is effective only during periods when the burner is off, there should be no significant difference compared to the baseline case. The AVD opens at the start of the burner cycle and the damper offers minimal obstruction to flow of combustion gases through the vent. When the damper is closed, there is very little draft up the vent which results in conservation of heat within the building. If hypothetical furnace efficiency were to be calculated on the basis of net heat energy added to the room divided by the equivalent heat content of the natural gas, then the tests conducted in this study would not reflect the true value of the AVD device. Provided the furnace is placed inside the building, whether in a heated or unheated space, air flow up the stack results in heat losses and causes infiltration of cold air into the building. In such cases, overall furnace performance should be judged on net heat energy supplied to the rooms.

As an exercise for comparative purposes, instead of subtracting heat losses through the vent for the baseline, Thrifty Vent and vent restrictor tests, the quantity of heat that would have been lost during burner off periods for the baseline case was instead <u>added</u> to the total heat energy supplied by the furnace for tests with the AVD. While the calculated furnace efficiencies are artifically high in the calculations, never the less comparative differences in efficiencies may be observed. Calculations of heat losses were made for both heated air and unheated air and added to the measured quantities of heat supplied during the AVD tests. Air flow rates were measured in the vent for the baseline case during burner off periods and an outdoor air temperature of $-6^{o}C$ was assumed.

The furnace efficiencies resulting from the calculations are presented in Table 4. Also shown is the percent improvement over the baseline case. As expected the most significant improvements took place for the short burner cycles and where the fan was simulating a high wind draft condition. Again it must be stressed that the values reported in Table 4 are for assumed conditions relating to infiltration air temperature and source.

All the devices were compared to a baseline condition where no further adjustments were made to the furnace. While this was judged a reasonable way to make comparisons, two of the devices are intended to be operated, or can be operated in conjunction with derating of the furnace. If the furnace should be derated, a baseline test should be performed to establish a basis of comparison. The plots of flue gas temperature for the Thrifty-Vent and the vent restrictor relative to the lack of increased temperature in the room air delivery system suggest that a furnace derating might prove to be a valid test. If the entire temperature curve can be displaced downward to a temperature range near the baseline by derating the furnace, then some further improvements in furnace efficiency might be gained if the room air delivery temperature remains about the same as for the baseline case.

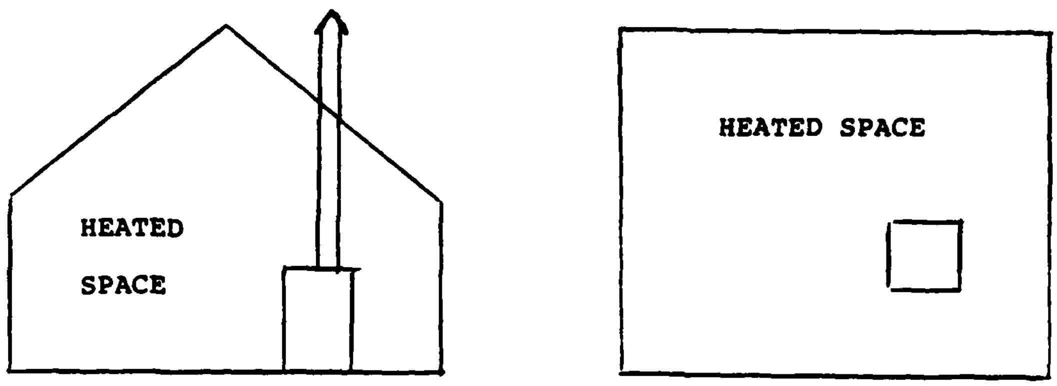
TABLE 4. ADJUSTED EFFICIENCIES FOR THE AUTOMATIC VENT DAMPER TO ACCOUNT FOR THE DECREASE IN AIR FLOW UP THE VENT STACK DURING BURNER OFF CYCLE. OUTSIDE AIR (INFILTRATION AIR) IS ASSUMED -6°C.

Device	Furnace room temp.	Stack fan	Duty cycle 10%	20%	30%	50%	75%	100%
Automatic vent damper adjusted for air flow during burner off cycle 76.7% overall	Heated	Off	68.8	76.5	77.1	82.8	78.1	80.7
		Low	67.5	75.0	76.4	77.6	77.3	79.6
		High	76.0	81.4	82.2	81.6	79.5	79.5
	Unheated	Off	65.2	76.9	78.5	78.6	78.8	80.2
		Low	69.7	75.6	77.4	77.1	76.6	78.4
		High	73.3	73.7	73.4	75.3	77.2	77.2
Automatic vent damper adjusted improvement in efficiency vs. baseline 4.6% overall	Heated	Off	5.4	4.3	0.5	0.4	1.1	3.6
		Low	12.4	2.7	1.6	1.1	1.1	3.6
		High	9.5	10.2	6.9	6.0	2.7	-0.7
	Unheated	Off	5.0	6.8	7.9	4.1	1.8	2.3
		Low	10.4	6.1	5.7	3.2	1.1	0.8
		High	18.8	7.8	4.2	3.2	3.0	2.3

PERFORMANCE OF THE DEVICES

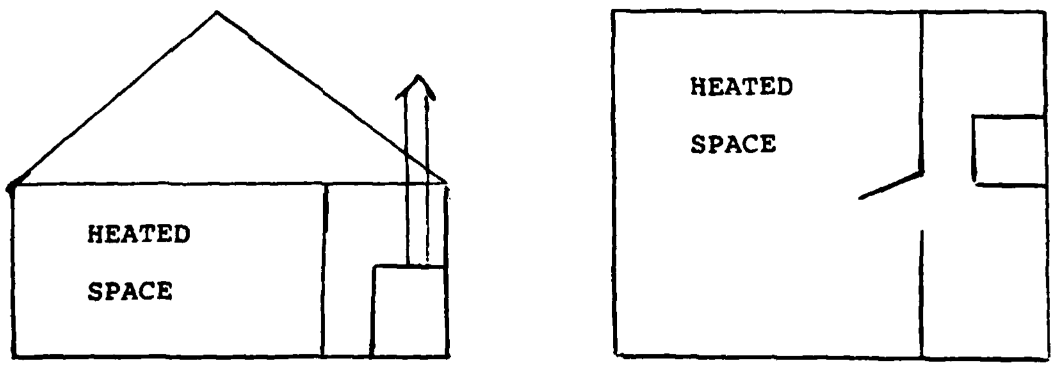
Application of the testing results to rational selection of one of the devices to achieve energy conservation in a gas fired furnace is a very complicated and confusing exercise. It is, perhaps, unfair to compare any of the devices to another one of the devices solely on the basis of the results of these tests. Each device has its own peculiar effect on the furnace and this effect is different under different circumstances. For example, the automatic vent damper was designed to be of benefit in reducing exfiltration air through the vent during the burner "off" cycle only. It was not designed nor was it anticipated to have an effect on the furnace during the "on" cycle. This was found to be true. The remaining two devices were claimed to have benefits during both the off and on cycles. This was also found to be true, but to varying degrees and under different circumstances. Each of the devices was found to be markedly affected in performance by furnace duty cycle, wind velocity, and location of the furnace in heated or unheated space. The widely varying results obtained under different conditions explain to some extent the wide disparity of results reported in the Literature Survey Report from Task Order 1.

The results of this present study should not be used as a basis for selecting a furnace vent device for energy conservation. The results, however, can be used as a guideline. As an example, Tables 5, 6, and 7 represent calculated, anticipated, annual energy cost savings based upon the vent flow rates, furnace efficiencies, and assuming a 2,800 hour heating demand season with a $2.00 per million BTU gas cost. The best performing device for each circumstance is represented in Table 8. Those results shown under "Heated Space" may be considered applicable to typical Army housing construction where the furnace is centrally located on the floor of slab-on-grade construction as shown in the following sketch.



Furnace located in heated space. All combustion and dilution air is conditioned air.

Those results shown under "Unheated Space" are somewhat applicable to barracks construction with a furnace room at one end. However, in this type of construction the vent may penetrate the roof or wall in unconditioned space and the outside (only) access door may be louvered. In such a case, the represented savings will be somewhat less. See the following sketch.



Furnace located in unheated space. All combustion and dilution air is replaced by conditioned air.

It is critical to keep in mind that these hypothetical annual savings were based upon test results of one furnace with one 5 inch vent in a laboratory test room. Furnace and vent interface with an actual building could have a very significant effect upon both the magnitude and comparison of results. Furnace size and vent diameter are two of a number of factors which would have a significant effect on the values and comparisons reported. Vent temperatures and flow rates achieved with the vent restrictor and Thrifty-Vent suggest that their greatest benefit would be achieved when used in combination with a careful derating of the furnace device. In fact, it has been reported that Thrifty-Vent derates furnaces as a general policy when they install the device. Public Service Company of Colorado has a field test program under way now in which the vent restrictor is used in combination with furnace derating.

Based upon actual field practice and the results of these tests, it is strongly recommended that a second series of tests be conducted in which several levels of furnace derating are undertaken and then tested with each of the vent devices included in this present study. In this way, the maximum benefit of the Thrifty-Vent and vent restricter may be disclosed.

TABLE 5. TOTAL ANNUAL ENERGY COST SAVINGS ATTRIBUTABLE TO AUTOMATIC VENT DAMPER INSTALLED ON 110,000 BTU FURNACE; 2800 HOUR HEATING SEASON, 30°F AVERAGE OUTSIDE TEMPERATURE; 5" DIAMETER VENT, \$2.00 PER 1,000,000 BTU.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	\$2.75	\$11.64	\$ 0.93	\$18,22
Medium, runs 50% of time	-\$0.53	\$19.82	\$14.59	\$12.60
High, runs 75% of time	\$6.73	\$14.16	\$ 0.02	\$14.60

TABLE 6. TOTAL ANNUAL ENERGY COST SAVINGS ATTRIBUTABLE TO THRIFTY VENT INSTALLED ON 110,000 BTU FURNACE ; 2800 HOUR HEATING SEASON, 30°F AVERAGE OUTSIDE TEMPERATURE, 5" DIAMETER VENT, $2.00 PER 1,000,000 BTU.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	$ 2.13	-$ 0.61	$ 1.40	$10.18
Medium, runs 50% of time	$ 0.82	$14.12	$21.99	$29.43
High, runs 75% of time	$29.28	$11.66	$26.87	$37.70

TABLE 7. TOTAL ANNUAL ENERGY COST SAVINGS ATTRIBUTABLE TO VENT RESTRICTOR INSTALLED ON 110,000 BTU FURNACE; 2800 HOUR HEATING SEASON, 30°F AVERAGE OUTSIDE TEMPERATURE, 5" DIAMETER VENT, $2.00 PER 1,000,000 BTU.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	$ 4.81	-$ 0.27	-$ 0.35	$11.35
Medium, runs 50% of time	$16.64	$ 3.33	$ 0.16	$13.15
High, runs 75% of time	$ 4.68	$ 0.41	$ 0.24	$17.42

TABLE 8. BEST PERFORMING DEVICE. NUMERICAL RATING EQUALS ANNUAL SAVINGS DIVIDED BY HIGHEST SAVINGS TIMES 100.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	Vent restrictor 13	Automatic vent damper 31	Thrifty Vent 4	Automatic vent damper 48
Medium, runs 50% of time	Thrifty Vent 2	Automatic vent damper 53	Thrifty Vent 58	Thrifty Vent 78
High, runs 75% of time	Thrifty Vent 78	Automatic vent damper 38	Thrifty Vent 71	Thrifty Vent 100

Safety considerations were not a subject of this study. However, changes in flow rates, vent temperatures, and heat exchanger temperatures achieved with both the vent restrictor and Thrifty-Vent raise certain questions concerning their effect on condensation and resultant corrosion as well as heat exchanger life. It is recommended that these data be carefully reviewed with the local utility and building inspector before installation of either type of device is undertaken. Following are some comments relative to each individual device beyond the original objective of this task order which was limited to furnace thermal efficiency.

Automatic Vent Damper

The automatic vent damper is designed to reduce significantly air flow up the vent during the furnace off cycle. It is not designed to have any major effect on furnace operation. This was found to be true. Table 9 displays the Btu saved by reduced vent exfiltration, and Table 10 those saved through improved furnace efficiency. These savings are carried into dollar savings in Tables 11 and 12. It is important to keep in mind that these tests were conducted in a closed laboratory condition with a 5 inch vent. Furnaces in real houses with air infiltration and an induced stack effect should benefit even more from the installation of an AVD. Also, larger vents may enhance the benefits to be achieved through the installation of an AVD. It appears that AVD's will be very cost effective in high wind areas; in large diameter vents, such as 8 inches; in houses having very high air infiltration rates. and in houses with greatly oversized furnaces operating on low duty cycles. It appears that AVD may not be cost effective in very low wind areas; in tight houses; and in circumstances where the furnace runs more than 50 percent of the time. It is anticipated that the final results of the American Gas Association's SHEIP program to be published sometime in 1980 will provide a good basis for calculating the cost effectiveness of AVD's under a variety of circumstances.

TABLE 9. ENERGY SAVED BY REDUCED VENT EXFILTRATION DURING HEATING SEASON USING AUTOMATIC VENT DAMPER, BTUH PER °F TEMPERATURE DIFFERENCE AND 5" DIAMETER VENT.

Furnace duty cycle	Furnace located in			
	Heated space		Unheated space	
	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	15.6	65.7	1.7	67.3
Medium, runs 50% of time	4.5	33.5	11.5	31.5
High, runs 75% of time	9.4	19.9	-6.1	13.6

TABLE 10. ENERGY SAVED THROUGH IMPROVED FURNACE EFFICIENCY USING AUTOMATIC VENT DAMPER; BTUH

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	-132	-550	99	561
Medium, runs 50% of time	-275	2200	2145	990
High, runs 75% of time	825	1732	248	2062

TABLE 11. ANNUAL ENERGY COST SAVINGS BY REDUCED VENT EXFILTRATION; 2800 HOUR HEATING SEASON; 30°F AVERAGE OUTSIDE TEMPERATURE; 5" DIAMETER VENT; $2.00 PER MM BTU GAS COST-USING AUTOMATIC VENT DAMPER

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	$3.49	$14.72	$0.38	$15.08
Medium, runs 50% of time	$1.01	$ 7.50	$2.58	$ 7.06
High, runs 75% of time	$2.11	$ 4.46	-$1.37	$ 3.05

TABLE 12. ANNUAL ENERGY COST SAVINGS THROUGH IMPROVED FURNACE EFFICIENCY USING AUTOMATIC VENT DAMPER 2800 HOUR HEATING SEASON; 110,000 BTU FURNACE, $2.00 per 1,000,000 BTU.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	-$0.74	-$ 3.08	$ 0.55	$ 3.14
Medium, runs 50% of time	-$1.54	$12.32	$12.01	$ 5.54
High, runs 75% of time	$4.62	$ 9.70	$ 1.39	$11.55

Vent Restrictor

Tables 13-16 depict the energy savings and cost effect attributable to the use of the vent restrictor. In general, it may be said that this device proved to be of little benefit under any of the circumstances in this test program. The vent restrictor did increase flue gas temperatures significantly. This was not accompanied by an increase in Btu delivered to the room supply air. Apparently the heat exchanger in the furnace was operating at maximum capability under normal operating conditions. This strongly suggests that the vent restrictor could be of significant value when used in combination with a derating of the burner. Theoretically, such a combination would maintain Btu delivered to the room supply air at a much lower Btu furnace input, and no change in design operating temperature of the heat exchanger. Such a change may well require altering the combustion and secondary air flow rates. This can only be determined through a series of tests.

Thrifty-Vent

Tables 17-20 depict the Btu and dollar savings achieved through Thrifty-Vent under the test conditions and hypothetical circumstance of heating season and gas costs. This was the most effective of the three devices. It was not within the scope of this Task Order to develop means for measuring cost effectiveness of Thrifty-Vent. No data are available concerning its installed costs on furnaces of various sizes and various parts of the country. If it is comparable in price to an AVD for example, it may, in fact, be cost effective under conditions of high wind and long duty cycles. This is especially true when the furnace is located in unheated space yet draws conditioned combustion and dilution air from within the building. Thrifty-Vent appears to be not cost effective in circumstance where the furnace has a very low duty cycle running, say, 10 percent of the time. These facts combined with the indication of increased heat exchanger temperature and reduced vent temperature strongly suggests, as with the vent restrictor, that Thrifty-Vent be evaluated in combination with a furnace derate. It appears that this combination might produce very significant energy savings. It is probably no coincidence that Thrifty-Vent was reported to derate a furnace as part of their normal installation.

TABLE 13. ENERGY SAVED BY REDUCED VENT EXFILTRATION DURING HEATING SEASON USING VENT RESTRICTOR BTUH PER °F TEMPERATURE DIFFERENCE AND 5" DIAMETER VENT.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	14.6	19.4	- 6.0	38.3
Medium, runs 50% of time	12.3	24.5	6.2	25.7
High, runs 75% of time	16.8	28.6	13.4	30.3

TABLE 14. ENERGY SAVED THROUGH IMPROVED FURNACE EFFICIENCY USING VENT RESTRICTOR; BTUH.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	275	- 825	176	495
Medium, runs 50% of time	-3465	- 385	-220	1320
High, runs 75% of time	165	-1072	-578	1898

TABLE 15. ANNUAL ENERGY COST SAVINGS BY REDUCED VENT EXFILTRATION; 2800 HOUR HEATING SEASON, 30°F AVERAGE OUTSIDE TEMPERATURE; 5" DIAMETER VENT; $2.00 PER MM BTU GAS COST- USING VENT RESTRICTOR

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	$3.27	$4.35	-$1,34	$8.58
Medium, runs 50% of time	$2,76	$5.49	$1.39	$5.76
High, runs 75% of time	$3.76	$6.41	$3.00	$6.79

TABLE 16. ANNUAL ENERGY COST SAVINGS THROUGH IMPROVED FURNACE EFFICIENCY USING VENT RESTRICTOR 2800 HOUR HEATING SEASON; 110,000 BTU FURNACE, $2.00 per 1,000,000 BTU.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	$ 1.54	-$4.62	-$0.99	$ 2.77
Medium, runs 50% of time	-$19.40	-$2.16	-$1.23	$ 7.39
High, runs 75% of time	$ 0.92	-$6.00	-$3.24	$10.63

TABLE 17. ENERGY SAVED BY REDUCED VENT EXFILTRATION DURING HEATING SEASON USING THRIFTY-VENT BTUH PER °F TEMPERATURE DIFFERENCE AND 5" DIAMETER VENT.

Furnace duty cycle	Furnace located in			
	Heated space		Unheated space	
	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	0,7	4,7	6,5	20,4
Medium, runs 50% of time	18.8	10.8	19,8	15,9
High, runs 75% of time	27.6	14.9	25.1	15.7

TABLE 18. ENERGY SAVED THROUGH IMPROVED FURNACE EFFICIENCY USING THRIFTY-VENT; BTUH

Furnace duty cycle	Furnace located in			
	Heated space		Unheated space	
	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	352	- 297	- 11	1001
Medium, runs 50% of time	- 605	2090	3135	4620
High, runs 75% of time	4125	1485	3795	6105

TABLE 19. ANNUAL ENERGY COST SAVINGS BY REDUCED VENT EXFILTRATION; 2800 HOUR HEATING SEASON; 30°F AVERAGE OUTSIDE TEMPERATURE; 5" DIAMETER VENT $2.00 PER MM BTU GAS COST-USING THRIFTY-VENT.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	\$0.16	\$1.05	\$1.46	\$4.57
Medium, runs 50% of time	\$4.21	\$2.42	\$4.43	\$3.56
High, runs 75% of time	\$6.18	\$3.34	\$5.62	\$3.51

TABLE 20. ANNUAL ENERGY COST SAVINGS THROUGH IMPROVED FURNACE EFFICIENCY USING THRIFTY VENT: 2800 HOUR HEATING SEASON; 110,000 BTU FURNACE; $2.00 per 1,000,000 BTU.

	Furnace located in			
	Heated space		Unheated space	
Furnace duty cycle	Low wind area	High wind area	Low wind area	High wind area
Low, runs 10% of time	$ 1.97	-$ 1.66	-$ 0.06	$ 5.61
Medium, runs 50% of time	-$ 3.39	$11.70	$17.56	$25.87
High, runs 75% of time	$23.10	$ 8.32	$21.25	$34.19

It was not within the scope of this study to investigate the safety aspects of the various devices. The marked increase in flue gas temperature and the decrease in upper vent temperature produced by Thrifty-Vent raised questions concerning heat exchanger life and condensation which should be addressed through local building officials and utilities. It is possible that when Thrifty-Vent is installed in combination with a furnace derate that these potential safety problems may disappear. This question could be addressed only through additional tests on the furnace used in this study. Several steps of derate should be investigated to address not only the safety questions but the maximum improvement in energy conservation which could be achieved with the Thrifty-Vent.

CONCLUSIONS

- An automatic vent damper was of value only during the furnace off cycle of these tests.
- The cost benefit of an automatic vent damper is greatest in high wind areas and on furnaces having low duty cycles.
- Thrifty-Vent was superior to a vent restrictor on an unmodified furance.
- Changes in temperatures and air flows strongly suggested that Thrifty-Vent and the vent restrictor may work best when combined with furnace derating.

RECOMMENDATIONS

It is recommended that the maximum potential benefit of Thrifty-Vent and a vent restrictor be measured by repeating the tests described in this report at several different levels of furnace derating.

It is recommended that the consideration of installing any furnace vent device be discussed with the local fuel supplier (utility), building officials and representatives of the furnace and vent device manufacturers. A building permit is likely to be required and the furnace manufacturer's warranty may be jeopardized.

APPENDIX A

THE TEST FACILITY AND INSTRUMENTATION

The test facility consisted of a "furnace room" 6 feet wide and 12 feet long and an instrumentation room the same size located immediately adjacent to the furnace room. The test rooms were located inside the Hydro-Machinery Laboratory space at the Engineering Research Center at the Foothills Campus of Colorado State University. The laboratory is a large enclosed building with a roof 22 feet off the floor and a total enclosed floor of 90 feet by 175 feet. This floor space is heated in the winter by radiant style heaters at the roof level with vent fans venting in the roof.

The "room air" for the test furnace was supplied from the laboratory space and heated air was delivered to the lab space at 8 feet above the floor and directed in an opposite direction from the cold air intake. The air temperature near the floor of the laboratory varied between 18° to $20^{\circ}C$ which is representative of return air temperature in a heated building.

A regulated natural gas supply line already existed within the building and was easily extended to the test rooms for operation of the furnace. The combustion vent was run through an office balcony located immediately above the test rooms and vented into a space near the "roof" inside the building. A vent fan, operating whenever the furnace was operating, then pulled the combustion gases outside the lab building. This arrangement enabled tests to be independent of random wind effects on the vent. To simulate a wind induced draft through the vent, a two speed vent fan was used. This allowed the effects of a wind induced draft to be duplicated consistently from run to run without being dependent on the actual wind. The vent fan was installed so that the flue gas flow would not be obstructed when the fan was not being used.

Combustion air for the furnace room was drawn in through a wall vent from the laboratory floor, well away from the "room air return" intake. To simulate a heated furnace room the furnace space was simply allowed to reach a uniform temperature that resulted from the operation of the furnace itself. For an unheated furnace room a package room air conditioner was installed in the furnace room to cool the furnace room down to about 10°C. During actual furnace operating cycles this temperature would gradually increase reaching a maximum for the 100 percent operating cycle of about 16°C.

The furnace used for the tests was a Lennox Model G1203-110 rated at 110,000 BTUH input. The furnace is a completely standard model just as shipped by the manufacturer. The unit included a four speed blower and a standing gas pilot light and all burner controls, fan switches and 24 volt control wiring. The furnace was purchased from a local distributor and delivered in the original factory carton.

Furnace installation was accomplished by a local heating contractor. Blower fan settings were placed on the "medium high" speed, which in many cases accommodated air conditioning systems installed in conjunction with the furnace unit. Fan thermostat settings were adjusted so that the fan would turn off at around 27°C (80°F). This setting was made to assist in smoothing out blower operation time through the various tests.

Air ducts to the furnace from the laboratory space were stubbed through the walls of the furnace room. All connections on the duct were sealed with a silicone sealant to prevent air leakage from or air infiltration into the "room air" system. The ducts were wrapped with 3-1/2 inches of fiber glass insulation with an R value of 11. Two Cambridge Filter Corporation air flow monitors Model FMS-D-14 X 14, consisting of pitot rakes and static pressure probes were installed, one in the cool air return duct and one in the warm air delivery duct. The duct air flow monitors, were calibrated at the Solar Energy Applications Laboratory using an ASME flow nozzle as the calibration standard. Each monitor consists of total and static pressure probe arrays which were separately manifolded to an outlet port. The differences between total and static pressures were measured with a Barocel electronic manometer. Average air velocity at the cross-section of the monitor, hence volumetric air flow rate and flow rate coefficients were determined from the calibrations. Figure A1 about 10°C. During actual furnace operating cycles this temperature would gradually increase reaching a maximum for the 100 percent operating cycle of about 16°C.

The furnace used for the tests was a Lennox Model G1203-110 rated at 110,000 BTUH input. The furnace is a completely standard model just as shipped by the manufacturer. The unit included a four speed blower and a standing gas pilot light and all burner controls, fan switches and 24 volt control wiring. The furnace was purchased from a local distributor and delivered in the original factory carton.

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Figure A-1. Test furnace (a), cold air return duct (b) and air flow monitor (c). The cold air return duct is attached to the base of the furnace. The air flow monitor, labeled (W102A), is seen in the duct against the wall. A thermocouple and a 3 junction thermopile (d) are mounted on the right side of the furnace cabinet for sensing furnace room air temperature and temperature difference with the stack gas temperature.

Instrumentation

The following data were taken for the analysis of each device on the vent stack:

1. Temperatures throughout the system

2. Air flow rates in the room air circuit

3. Vent flow rates during and before burner operation

4. Wet and dry bulb temperature

5. Gas consumption during each run

6. Electric power to the blower motor

Temperatures

Copper-constantan (Type T) thermocouples were used to monitor temperatures throughout the system during testing. Figures A2 and A3 are system schematics giving relative positions of temperature sensors. Thermocouples, designated by TC on the figures were used to measure temperatures relative to ice point ($0^{o}C$). The ice point reference cell, see Figure A4, provides a constant reference temperature of $0^{o}C$, in which one junction of every thermocouple circuit is placed to provide the reference temperature.

The thermocouples were formed from type T thermocouple wire manufactured to special limits of purity. For the vent positions at a high temperature, teflon insulated thermocouple wire was used and a ceramic thermal coating was applied over the exposed thermocouple to protect it from possible deterioration. The "special limits of error" (3/8%) wire used results in thermocouples of more uniform and predictable characteristics. Junctions were welded bead and measures were taken to insure a uniform and small size of the weld bead to reduce response time.

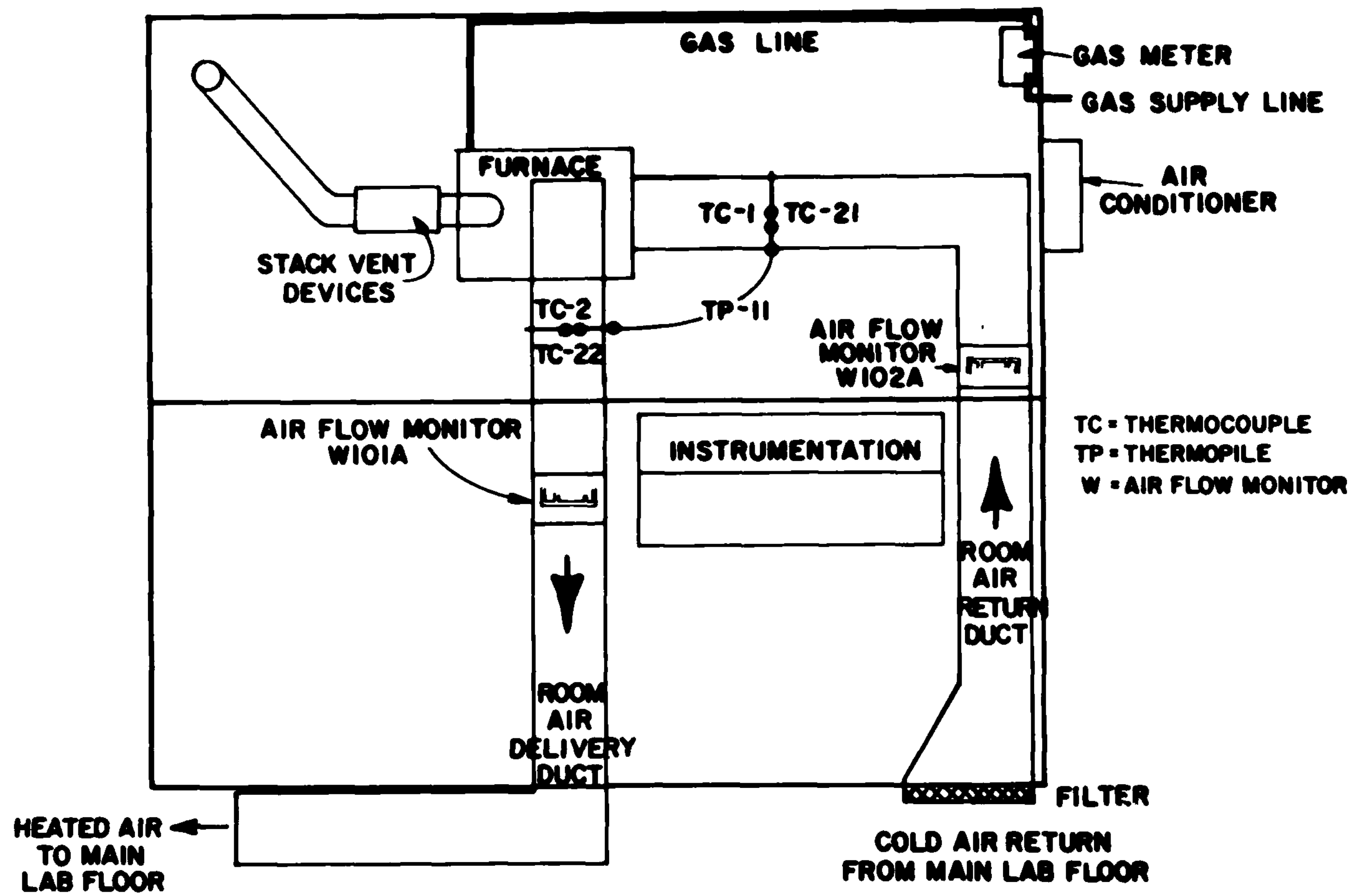

Figure A-2. General layout of the test facility and location of temperature sensors.

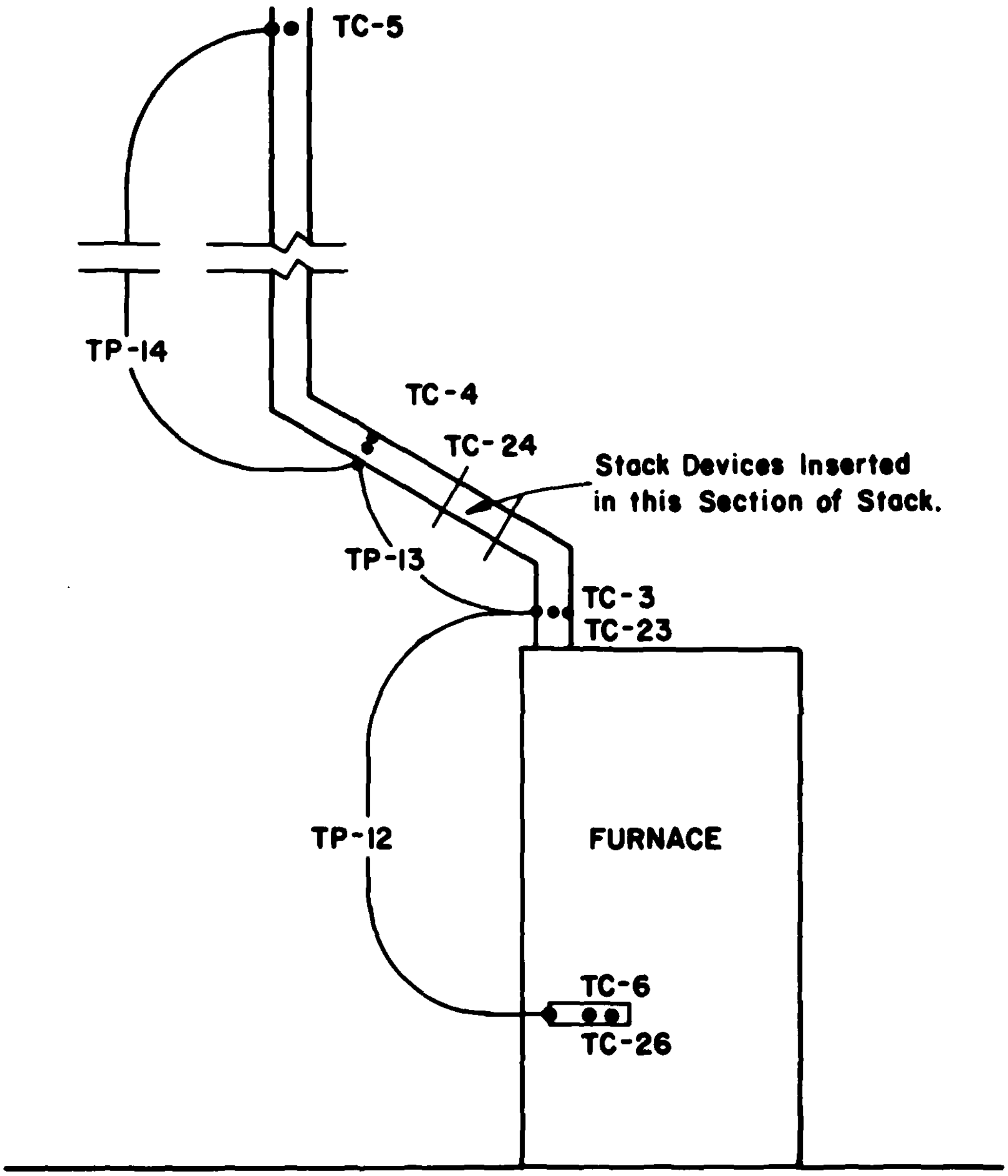

Figure A-3. Location of temperature sensors in the vent. (TC indicates a themocouple, TP indicates a thermopile)

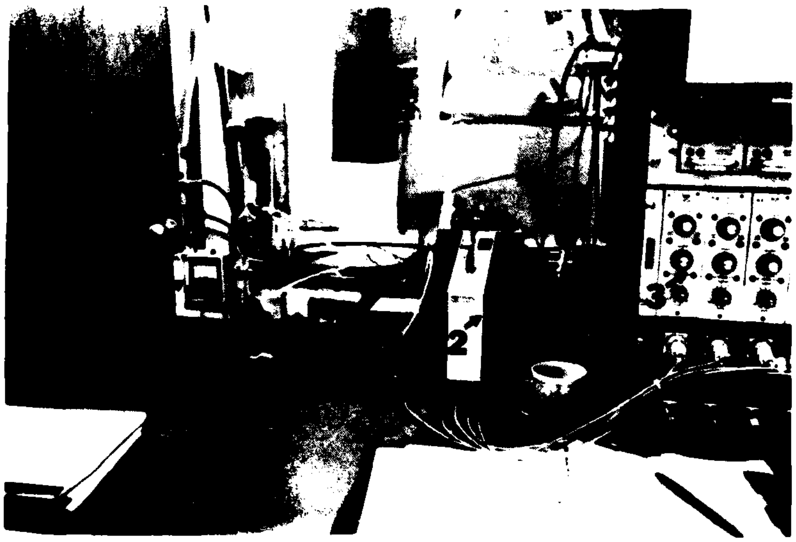

Figure A-4. The micro-manometer for measuring duct air flow is at the left, in the center is the ice point temperature reference for thermocouple junctions while to the right are the D.C. amplifiers.

Thermopiles, which are made up of individual thermocouple pairs wired in series, were used to measure all temperature differences. The thermopile circuits do not measure absolute temperature relative to some reference but rather, provide a measurement of temperature increase or decrease (delta) across the circuit points relative to either end of the circuit. For example, a nine junction thermopile was used to measure differences in air temperature across the furnace, designated as TP-11 in Figure A2. This thermopile detects the increase in temperature across the furnace between the cool air in and the warm air delivery.

Figure A5 is a view of the thermopile mounting rack in the cool air return duct. Each of the nine thermocouples making up the thermopile was placed at the centroid of an equal area in the cross section. In this way the thermopile provides an average of the air temperature through the duct.

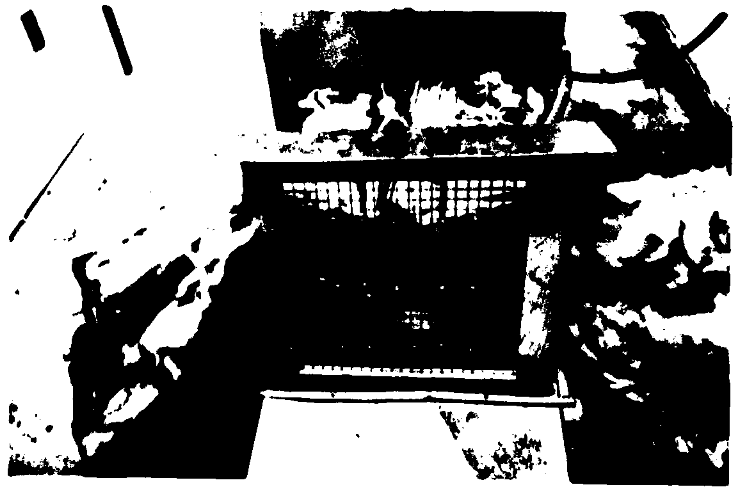

Figure A-5. Thermopile mounting rack for the air ducts to the furnace. Each junction is located at the center of an equal area segment of the duct cross section. A thermocouple is mounted at the center of the duct.

Data Acquisition System

To collect the temperature data over the 30 minute duty cycle with enough data points to make significant energy flow calculations a digital data acquisition system with magnetic tape recorder was used. The system used was a Doric Digitrend Model 210 linked with a Kennedy Incremental digital tape recorder. In a continuous scan mode the Doric sampled all temperatures and thermopiles approximately every 1.5 seconds and the data was recorded, via the Kennedy, on magnetic tape. Three data channels contained non-temperature data, the time of day being one and a voltage signal indication of thermostat on and blower on being the other two. Figure A6 is a view of the digital data acquisition system.

Air Flow Rates

To compute the energy flux from the furnace the air flow rates in the room air duct system during blower operations are required. Two Cambridge Filter Corporation duct air flow monitors were used for this purpose. A pressure differential at the air flow monitor is detected and the magnitude of that pressure differential is directly related to the air flow in the duct. To measure that pressure differential a Dwyer "Microtector" micro manometer was used. These readings were taken manually during the blower on period of each run and recorded on a data sheet for that run. Both the cold air return and warm air delivery ducts were monitored for air flow. Outflow rates were always greater than measured inflow rates to the furnace because of air leakage to the blower through the furnace cabinet.

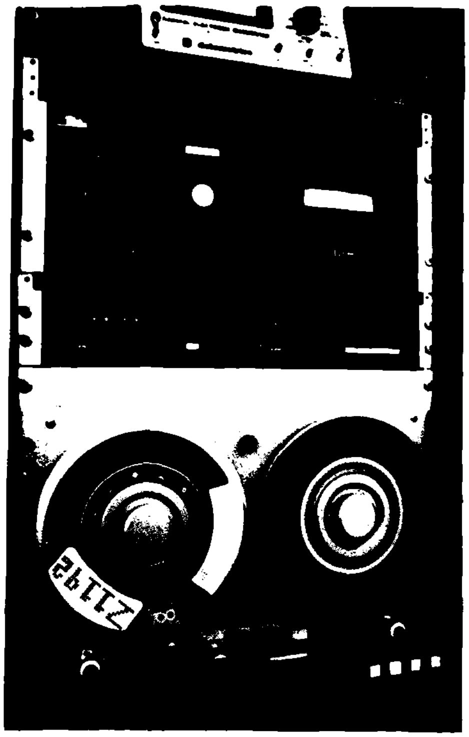

Figure A-6. The digital data acquisition system. The Barocel electronic manometer for measuring vent flow rates is seen at the top. During test runs the digital system scanned at a rate of once every 1.5 seconds producing 1200 points for each data channel for each test. There were 10 channels to the data acquisition system.

Vent Flow Rates

Vent flow rates were measured both before and during the burner cycle to determine the effect of each device on the draft during burner on and burner off periods. The measurements are important in estimating the overall energy savings attributable to each device.

Vent flow measurements were made with pitot tube at a point mid-way up the straight vent section (about 8 feet above the furnace). The pressure difference sensed by the pitot tube was relatively small when the burner was off and a sensitive Datametrics Corporation Barocel Electronic Manometer Model 1173 was used. The pitot tube hooked directly to the "Barocel" pressure sensing element through hard plastic tubing. Readings of vent flow ratio were taken manually on the Barocel analog meter during each test and noted on the data sheets.

Wet and Dry Bulb Temperatures

A Taylor Model 1328 sling psychrometer was used to measure dry bulb and wet bulb temepratures, both in laboratory space and in the furnace room. Readings were taken for each test run and manually recorded on the data sheets.

Gas Consumption

Gas consumption during each test run was obtained with a Sprague Model 175 CFH-Ser. No. S-2482095, gas meter which measures total cubic feet of gas used.

Gas meter readings combined with temperature, barometric pressure and gas quality provided by the local utility were used to determine energy content of the gas provided to the furnace.

Blower Power

The electric energy to the blower was measured with a power watt meter. This value was manually entered in the data sheets for each run. While parasitic power should be subtracted from the heat supplied by the burner, blower energy consumed was not large enough to have a significant effect on the comparisons of devices.

Test Procedure

The testing matrix for each device was as follows:

Device Under Test	Furnace Room	Vent Fan	Burner on % Duty Cycle
Baseline	Heated	Off	10, 20, 30, 50, 75, 100
Thrifty-Vent, Vent Restrictor, AVD		Low	10, 20, 30, 50, 75, 100
		High	10, 20, 30, 50, 75, 100
	Unheated	Off	10, 20, 30, 50, 75, 100
		Low	10, 20, 30, 50, 75, 100
		High	10, 20, 30, 50, 75, 100

Testing was scheduled so that one complete set of duty cycles, i.e., 10 through 100 percent, for a single test situation could be completed in one day. The duty cycle was selected to be 30 minutes and indicates the percent of time within the 30 minute period that the burner is "on". The duty cycle begins when the thermostat activates turning the burner on. For the test series a clock timer was used as "thermostat" contacts so that burner on times could be easily controlled and duplicated from test to test.

Each 30 minute test sequence included a 1 hour (2 complete duty cycles) pre-conditioning period with the burner on time during this pre-conditioning the same as that under test. The duty cycle during which data were collected was in effect the third cycle of the test set. Testing proceeded during the day from 10 percent to 100 percent in that order.

Baseline

A complete matrix of tests looking at the furnace operating into a normal 5 inch diameter vent would serve as the "baseline" against which energy saving vent devices would be compared. After the baseline tests had been completed, no changes were made to the furnace.

"Thrifty-Vent"

The first of the vent devices to be tested was a "Thrifty-Vent" model no. 5AR extended draft hood manufactured by Thrifty-Vent Inc., Salem, Illinois. This vent acts as a low gravity damper and gas separator, separating cooler stack gases and holding the hotter stack gases in the inlet compartment, see Figure A7. The "Thrifty-Vent" was installed in accordance with dimensions specified by the manufacturer and by a certified installer. The Thrifty-Vent under test was installed by the Denver area distributor.

The existing draft hood on the furnace was completely sealed as in Figure A8, and the open bottom of the Thrifty-Vent now served this function. Figure A9 shows the Thrifty-Vent installed on the test furnace. The bottom of the Thrifty-Vent must be on a line with the top of the now closed off furnace draft hood. This could be a serious limitation where overhead space in the furnace room is limited. The overall height of the test furnace Thrifty-Vent combination was 86 inches.

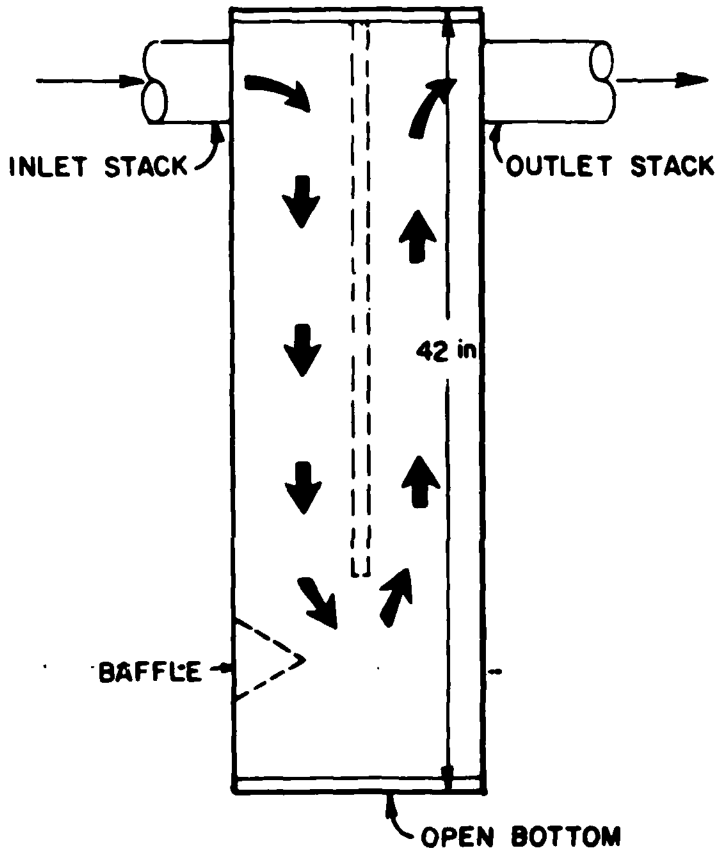

Figure A-7. Diagram of the thrifty vent showing how the stack gas is directed down and around a baffle to retain hot combustion products in the heat exchanger.

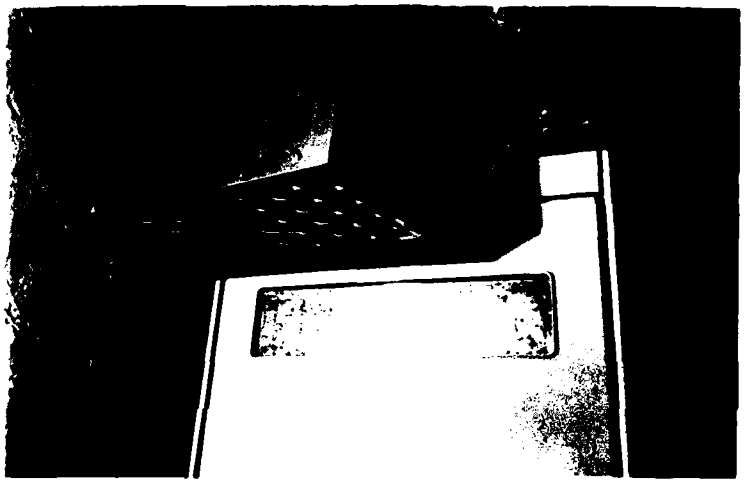

Figure A-8. View from underneath showing the open bottom of the Thrifty-Vent extended draft hood. Note the draft hood on the furnace has been sealed.

Figure A-9. Overall view of the "Thrifty-Vent"extended draft hood as installed in the test furnace vent.

Automatic Vent Damper (AVD)

The automatic vent damper is a motor driven damper which closes to block the vent when the burner is not being fired.

The unit consists of a motor, a relay and two cam activated microswitches in a control box mounted outside the vent. See Figures A10 and A11. The thermostat signal is wired into the AVD rather than to the burner control circuit. The burner control is in turn wired to a set of relay contacts in the AVD. When the thermostat signals for heat the AVD is activated and begins opening. When the AVD is fully open the microswitch is contacted by a cam and the relay is energized turning the burner on and stopping damper rotation.

There was a small problem during shut down. The burner control supplied with the test furnace had a delay between the time it is activated or deactivated electrically until it actually opens or closes the gas supply to the burner. Therefore, although the electrical signal to shut down is given when the thermostat contacts open, the damper is partially closed while the burner is still operating. This overlap consists of 10 seconds or so since the damper is driven through a gear drive that rotates it relatively slowly into position.

The automatic vent damper used was the Johnson Controls Corp., - Penn Division, Series Q-16 "Mizer", Goshen, Indiana. This unit is designed for installation only with an IID which may account for the small problem described here. It was not within the scope of this study to include testing of an IID which is an "intermittent ignition device". This usually is an electric spark ignitor which takes the place of the standing gas pilot light.

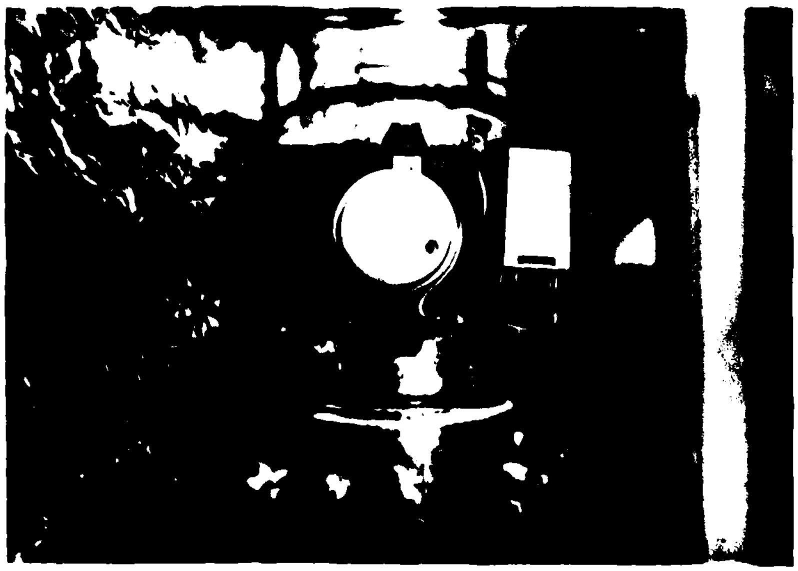

Figure A-10. View of the automatic vent damper with cover removed. To the left under incoming wiring is cam activated microswitch, center is the clock motor drive and to the left is the relay containing burner contacts and clock motor interrupt contacts.

Figure A-11. Automatic vent damper as installed in test furnace vent. Note the thermocouple and thermopile junctions enter the vent just below the AVD.

Vent Restrictor

The vent restrictor is the simplest of the devices tested, at least in construction and installation. It is essentially an orifice that slides into a cut in the vent that creates a restriction to the flow in the vent. A simpler version is a flat rectangular plate with a square edge which is simply slipped into the vent to create the partial closure of the vent. In either case the amount of restriction created to effectively block hot flue gas but not cause gas spillage at the draft hood must be determined for each installation.

A general procedure, which was followed for the test furnace, is to perform the adjustments on a warm day outdoors (21°C at the vent exhaust) so that natural vent draft is minimal. Beginning with a large diameter orifice plate, and with the burner on, (or with the vent restrictor removed) smaller and smaller diameter orifices are inserted in the vent in 1/4 inch diameter increments until gas spillage just starts at the draft hood. At the point of spillage the next larger size diameter plate is permanently installed. The test system had spillage just starting with a 3 inch diameter orifice in place and so a 3-1/4-inch orifice was installed for the test.

The vent restrictors used in the tests were of the centered orifice type, see Figures A12 and A13. These devices are typically made up locally by sheet metal workers and are not available directly from a manufacturer. Public Service Co. of Colorado provided the vent restrictors for the test furnace.

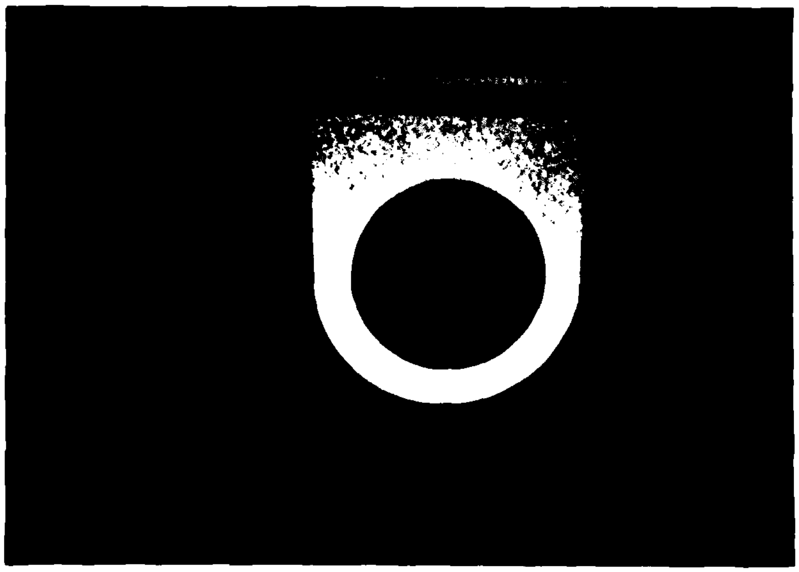

Figure A-12. Typical orifice vent restrictor used in the vent. The outer diameter of the flat piece just fits inside the vent diameter.

Figure A-13. The vent restrictor as installed in the test furnace vent. A 3¼ inch diameter opening was used in the test furnace.

APPENDIX B

OUTLINE OF KEY DATA FOR EACH DEVICE

TABLE B-1. OUTLINE OF DATA FOR BASELINE

Burner Duty Cycle		Heated Room			Cold Room		
		Stack Fan			Stack Fan		
		No Fan	Lo Fan	Hi Fan	No Fan	Lo Fan	Hi Fan
10%	Gas Input BTU's	5229.3	4990.4	5350.7	5147.0	5400.0	5283.6
	Return Air Monitor BTU's	3097.0	3237.9	3330.7	2905.2	2873.1	2634.6
	Room Air Delivery BTU's	3315.1	3468.2	3559.2	3098.8	3199.8	2880.1
	Return Air Efficiency	59.2	64.9	62.3	56.4	53.2	49.9
	Room Air Efficiency	63.4	69.5	66.5	60.2	59.3	54.5
	Stack Flow Rate*-Burner Off	23.70	78.48	79.08	13.59	74.39	76.47
	Stack Flow Rate*-Burner Onn	76.20	94.58	98.71	73.17	100.74	102.98
	Thermocouple Δ BTU Check	3529.7	3665.3	3538.7	3278.9	561.3	1442.7
20%	Gas Input BTU's	10361.0	10471.1	10558.1	10276.3	10456.2	10315.2
	Return Air Monitor BTU's	7036.6	7039.0	7087.0	6825.4	6440.5	6054.4
	Room Air Delivery BTU's	7479.6	7567.5	7514.9	7260.2	7268.9	6793.5
	Return Air Efficiency	67.9	67.2	67.1	66.4	61.6	58.7
	Room Air Efficiency	72.2	72.3	71.2	70.1	69.5	65.9
	Stack Flow Rate*-Burner Off	23.74	78.35	76.80	19.15	74.01	76.31
	Stack Flow Rate*-Burner On	80.01	95.53	103.45	74.19	102.13	104.55
	Thermocouple Δ BTU Check	7927.6	7960.6	7869.7	7546.0	2213.5	2588.3
30%	Gas Input BTU's	15503.2	15496.5	15613.2	15660.8	15446.3	15631.7
	Return Air Monitor BTU's	11185.8	10760.2	11142.9	10491.8	10052.7	9682.9
	Room Air Delivery BTU's	11867.3	11590.0	11748.3	11052.0	11072.4	10822.5
	Return Air Efficiency	72.2	69.4	71.4	67.0	65.1	61.9
	Room Air Efficiency	76.6	74.8	75.3	70.6	71.7	69.2
	Stack Flow Rate*-Burner Off	23.86	77.27	76.89	23.50	72.81	71.26
	Stack Flow Rate*-Burner On	77.98	101.47	103.76	77.98	98.47	101.90
	Thermocouple Δ BTU Check	12195.4	12357.4	12053.3	11671.1	3753.4	4761.6
50%	Gas Input BTU's	25245.9	25819.1	22556.0	25624.5	26011.8	2589.73
	Return Air Monitor BTU's	19614.3	18605.7	18342.1	17729.8	17181.6	17114.5
	Room Air Delivery BTU's	20800.0	19759.6	19309.1	19078.8	19215.9	18660.3
	Return Air Efficiency	77.7	72.1	71.2	69.2	66.1	66.1
	Room Air Efficiency	82.4	76.5	75.6	74.5	73.9	72.1
	Stack Flow Rate*-Burner Off	19.58	77.28	77.27	19.19	75.45	73.90
	Stack Flow Rate*-Burner On	78.35	103.36	103.14	74.60	101.52	102.94
	Thermocouple Δ Btu Check	20529.4	21132.2	20486.5	19956.4	8392.7	8630.5
75%	Gas Input BTU's	38136.9	38465.4	38487.9	38317.4	38462.0	38508.5
	Return Air Monitor BTU's	27227.8	27246.4	28161.7	27970.7	26009.7	25674.6
	Return Air Delivery BTU's	29346.6	29306.3	29571.1	29508.1	29041.2	28580.4
	Return Air Efficiency	71.4	70.8	73.2	73.0	67.6	66.7
	Room Air Efficiency	77.0	76.2	76.8	77.0	75.5	74.2
	Stack Flow Rate*-Burner Off	19.63	76.11	79.65	13.59	71.67	74.11
	Stack Flow Rate*-Burner On	76.10	100.58	102.38	69.45	97.92	98.74
	Thermocouple Δ BTU Check	31316.0	31215.5	31358.9	29961.8	13240.2	13645.9
100%	Gas Input BTU's	50488.6	51206.1	50440.7	50644.1	50953.7	51000.7
	Return Air Monitor BTU's	36273.5	36299.6	41439.9	37835.0	35679.2	34424.0
	Room Air Delivery BTU's	38900.7	38916.8	40460.0	39448.1	39561.7	38229.3
	Return Air Efficiency	71.8	70.9	82.2	74.7	70.0	67.5
	Room Air Efficiency	77.1	76.0	80.2	77.9	77.6	75.0
	Stack Flow Rate*-Burner Off	-------	-------	-------	-------	-------	-------
	Stack Flow Rate*-Burner On	74.81	97.77	99.78	74.78	98.05	100.69
	Thermocouple Δ BTU Check	41226.6	41493.7	46214.2	41911.5	21785.0	19483.7

*Flow rate is in ft^3/min

TABLE B-2. OUTLINE OF DATA FOR THRIFTY-VENT

Burner Duty Cycle		Heated Room Stack Fan No Fan	Heated Room Stack Fan Lo Fan	Heated Room Stack Fan Hi Fan	Cold Room Stack Fan No Fan	Cold Room Stack Fan Lo Fan	Cold Room Stack Fan Hi Fan
10%	Gas Input BTU's	5535.9	5177.8	5335.6	5466.1	5396.0	5375.7
	Return Air Monitor BTU's	3388.7	3088.3	3051.7	2875.9	2897.0	3076.5
	Room Air Delivery BTU's	3684.5	3439.5	3401.3	3285.6	3298.6	3420.9
	Return Air Efficiency	61.2	59.7	57.2	52.6	53.7	57.2
	Room Air Efficiency	66.6	66.4	63.8	60.1	61.1	63.6
	Stack Flow Rate*-Burner Off	27.35	80.89	81.97	12.02	80.94	79.42
	Stack Flow Rate*-Burner On	52.91	87.62	88.06	50.79	89.58	88.17
	Thermocouple Δ BTU Check	3664.9	3495.7	3515.4	3294.9	3288.7	3552.6
20%	Gas Input BTU's	10254.1	10398.9	10514.0	10230.7	10420.6	10365.4
	Return Air Monitor BTU's	7109.2	7253.0	7296.6	6744.8	6898.3	7012.6
	Room Air Delivery BTU's	7855.6	8197.9	8034.4	7474.7	7643.6	7591.5
	Return Air Efficiency	69.3	69.8	69.4	65.9	66.2	67.7
	Room Air Efficiency	76.6	78.8	76.4	73.1	73.4	73.2
	Stack Flow Rate*-Burner Off	19.42	81.53	80.82	16.69	77.62	80.45
	Stack Flow Rate*-Burner On	51.40	90.38	90.77	59.51	89.83	90.06
	Thermocouple Δ BTU Check	7979.0	8485.3	8138.6	7638.0	7532.5	7901.0
30%	Gas Input BTU's	15501.7	15636.9	15506.6	15393.9	15544.3	15422.2
	Return Air Monitor BTU's	11316.4	11421.3	10930.9	10752.0	10737.1	10880.8
	Room Air Delivery BTU's	12451.8	12559.9	12048.1	11876.2	11861.0	11737.0
	Return Air Efficiency	73.0	73.0	70.5	69.9	69.1	70.6
	Room Air Efficiency	80.3	80.3	77.7	77.2	76.3	76.1
	Stack Flow Rate*-Burner Off	13.75	80.50	82.19	21.53	74.89	76.64
	Stack Flow Rate*-Burner On	54.11	89.47	87.81	51.01	87.36	90.16
	Thermocouple Δ BTU Check	12438.7	12973.8	12401.6	12144.8	11799.5	12048.0
50%	Gas Input BTU's	25609.2	25565.2	25517.8	25439.9	25700.7	25921.2
	Return Air Monitor BTU's	19346.1	18767.3	18452.7	18555.6	18607.1	19057.1
	Room Air Delivery BTU's	20874.9	20638.2	20267.3	20406.2	20478.7	20862.0
	Return Air Efficiency	75.5	73.4	72.3	72.9	72.4	73.5
	Room Air Efficiency	81.3	80.7	79.4	80.2	79.7	80.5
	Stack Flow Rate*-Burner Off	16.89	79.31	81.43	13.64	76.16	80.53
	Stack Flow Rate*-Burner On	51.49	88.39	92.28	58.01	88.52	93.15
	Thermocouple Δ BTU Check	20587.0	21240.4	21091.2	20842.3	20279.4	21635.1
75%	Gas Input BTU's	38300.6	38604.1	39641.4	38088.9	38172.1	38409.0
	Return Air Monitor BTU's	28558.8	28823.3	28393.7	38309.0	29055.1	29263.4
	Room Air Delivery BTU's	31403.2	31607.1	31156.8	31076.9	31168.3	31339.7
	Return Air Efficiency	74.6	74.7	71.6	74.3	76.1	76.2
	Room Air Efficiency	82.0	81.9	78.6	81.6	81.7	81.6
	Stack Flow Rate*-Burner Off	N.A.	81.55	80.59	13.71	78.06	81.50
	Stack Flow Rate*-Burner On	54.32	89.29	90.65	48.49	86.03	91.27
	Thermocouple Δ BTU Check	31347.8	32603.5	32312.3	31927.8	31545.4	32575.0
100%	Gas Input BTU's	50661.1	50795.5	50485.2	50108.9	50426.3	50584.9
	Return Air Monitor BTU's	37248.9	38033.5	38599.7	37107.9	38227.8	38624.2
	Room Air Delivery BTU's	40794.4	41561.6	41194.6	41551.4	41803.3	42522.5
	Return Air Efficiency	73.5	74.9	76.5	74.1	75.8	76.4
	Room Air Efficiency	80.5	81.8	81.6	82.9	82.9	84.1
	Stack Flow Rate*-Bruner Off	-------	-------	-------	-------	-------	-------
	Stack Flow Rate*-Burner On	55.33	89.25	92.01	58.26	86.14	90.29
	Thermocouple Δ BTU Check	41644.5	42947.1	43028.9	42848.9	42589.6	44081.4

*Flow rate is in ft^3/min

TABLE B-3. OUTLINE OF DATA FOR AUTOMATIC VENT DAMPER

Burner Duty Cycle		Heated Room			Cold Room		
		Stack Fan			Stack Fan		
		No Fan	Lo Fan	Hi Fan	No Fan	Lo Fan	Hi Fan
10%	Gas Input BTU's	4817.8	4963.5	4991.9	4865.1	4837.6	4816.1
	Return Air Monitor BTU's	2780.7	2607.0	2881.6	2840.9	2635.7	2689.4
	Room Air Delivery BTU's	2995.6	2733.1	3069.9	2970.6	2810.4	2870.0
	Return Air Efficiency	57.7	52.5	57.7	58.4	54.5	55.8
	Room Air Efficiency	62.2	55.1	61.5	61.1	58.1	59.6
	Stack Flow Rate*-Burner Off	16.74	38.40	38.71	13.60	39.44	43.67
	Stack Flow Rate*-Burner On	42.23	98.83	100.58	66.64	98.49	100.90
	Thermocouple Δ BTU Check	3116.7	3316.2	3184.8	2973.6	2863.9	2901.3
20%	Gas Input BTU's	9935.7	9909.7	9819.9	9795.4	9863.9	9922.9
	Return Air Monitor BTU's	6669.1	6514.9	6917.6	7004.5	6584.5	6399.0
	Room Air Delivery BTU's	7373.7	6850.1	7328.1	7266.5	6981.3	6794.3
	Return Air Efficiency	67.1	65.7	70.4	71.5	66.8	64.5
	Room Air Efficiency	74.2	69.1	74.6	74.2	70.8	68.5
	Stack Flow Rate*-Burner Off	13.66	38.45	38.73	19.28	35.71	41.57
	Stack Flow Rate*-Burner On	61.09	99.95	100.72	76.02	99.39	100.20
	Thermocouple Δ BTU Check	7471.2	8126.0	7583.1	7305.1	7057.7	7108.9
30%	Gas Input BTU's	14815.1	15072.9	14804.4	15121.1	15258.1	14956.3
	Return Air Monitor BTU's	10370.5	10391.0	10933.2	10986.9	11000.9	10140.0
	Room Air Delivery BTU's	11212.4	10921.1	11541.1	11625.5	11351.0	10510.0
	Return Air Efficiency	70.0	68.9	73.9	77.7	72.1	67.6
	Room Air Efficiency	75.7	72.5	78.0	76.9	74.4	70.3
	Stack Flow Rate*-Burner Off	13.67	38.45	38.85	19.27	40.60	40.35
	Stack Flow Rate*-Burner On	77.88	97.78	103.86	68.32	98.81	[illegible]
	Thermocouple Δ BTU	11562.1	12911.2	12017.1	11713.2	11984.5	[illegible]
50%	Gas Input BTU's	23942.6	25213.3	25021.5	25331.6	25382.6	25468.4
	Return Air Monitor BTU's	18202.0	18236.8	18960.3	18953.6	18355.0	18224.3
	Room Air Delivery BTU's	19602.9	19083.9	19924.7	19847.1	19199.2	18823.4
	Return Air Efficiency	76.0	72.3	75.8	74.8	72.2	71.6
	Room Air Efficiency	81.9	75.7	79.6	78.35	75.6	73.9
	Stack Flow Rate*-Burner Off	19.39	38.90	39.06	13.66	40.62	40.52
	Stack Flow Rate*-Burner On	71.23	101.15	101.59	68.39	99.66	97.58
	Thermocouple Δ BTU Check	20357.9	22372.4	20894.6	20306.1	19872.3	19739.8
75%	Gas Input BTU's	38179.8	38561.8	37752.1	37860.9	38888.7	37999.3
	Return Air Monitor BTU's	27041.9	38896.7	28355.4	28770.8	28135.5	27725.3
	Room Air Delivery BTU's	29743.2	29586.6	29785.9	29778.8	29601.1	29135.4
	Return Air Efficiency	70.8	74.9	75.1	76.0	72.4	73.0
	Room Air Efficiency	78.0	76.7	78.9	77.3	76.1	76.7
	Stack Flow Rate*-Burner Off	13.75	36.49	33.93	19.30	38.36	40.68
	Stack Flow Rate*-Burner On	70.07	98.53	101.80	73.61	96.05	99.75
	Thermocouple Δ BTU Check	30966.2	33111.6	31269.9	29899.3	30289.5	30804.1
100%	Gas Input BTU's	50883.6	50774.4	50332.5	50517.6	51138.5	51157.9
	Return Air Monitor BTU's	36640.8	37990.8	37380.5	38963.2	39385.9	37703.1
	Room Air Delivery BTU's	41040.1	40415.2	40017.9	40535.3	40112.5	39470.3
	Return Air Efficiency	72.0	74.8	74.3	77.1	77.0	73.7
	Room Air Efficiency	80.7	79.6	79.5	80.2	78.4	77.2
	Stack Flow Rate*-Burner Off	-------	-------	-------	-------	-------	-------
	Stack Flow Rate*-Burner On	71.48	99.56	100.93	64.19	99.79	100.59
	Thermocouple Δ BTU Check	42767.4	45224.0	42031.9	41775.4	42301.4	41790.1

*Flow rate is in ft^3/min

TABLE B-4. OUTLINE OF DATA FOR VENT REACTOR

Burner Duty Cycle		Heated Room			Cold Room		
		Stack Fan			Stack Fan		
		No Fan	Lo Fan	Hi Fan	No Fan	Lo Fan	Hi Fan
10%	Gas Input BTU's	5238.4	5331.0	5238.8	5453.2	5422.3	5413.8
	Return Air Monitor BTU's	3223.7	3128.1	2905.9	3151.7	3041.2	2986.7
	Room Air Delivery BTU's	3449.7	3335.8	3091.1	3370.4	3252.5	3191.6
	Return Air Efficiency	61.5	58.7	55.5	57.8	56.1	55.2
	Room Air Efficiency	65.9	62.6	59.0	61.8	60.0	59.0
	Stack Flow Rate*-Burner Off	16.67	64.11	66.31	19.11	64.65	63.41
	Stack Flow Rate*-Burner On	60.85	83.13	83.41	63.36	80.91	76.59
	Thermocouple Δ BTU Check	3488.8	3543.2	3380.1	3598.9	3523.5	3434.9
20%	Gas Input BTU's	10323.7	10233.9	10355.0	10431.9	10454.2	10374.0
	Return Air Monitor BTU's	7004.8	7142.5	7197.9	6789.9	6675.4	6381.0
	Room Air Delivery BTU's	7399.5	7576.7	7618.5	7202.8	7080.3	6741.0
	Return Air Efficiency	67.9	69.8	69.5	65.1	63.9	61.5
	Room Air Efficiency	71.7	74.0	73.6	69.1	67.7	65.0
	Stack Flow Rate*-Burner Off	10.67	65.52	66.38	19.06	65.84	60.46
	Stack Flow Rate*-Burner On	63.18	80.88	81.38	66.10	80.77	81.28
	Thermocouple Δ BTU Check	7678.7	7752.6	7986.5	7675.1	7623.6	7318.8
30%	Gas Input BTU's	15450.2	15344.3	15515.1	15498.4	15506.3	15551.6
	Return Air Monitor BTU's	11077.6	10474.6	10506.4	11060.1	10581.1	10431.4
	Room Air Delivery BTU's	11708.1	11043.4	11053.2	11700.7	11189.6	10699.5
	Return Air Efficiency	71.7	68.3	67.7	71.4	68.2	67.1
	Room Air Efficiency	75.8	72.0	71.2	75.5	72.2	68.8
	Stack Flow Rate*-Burner Off	19.31	65.54	65.07	19.06	61.69	60.35
	Stack Flow Rate*-Burner On	61.69	80.88	82.54	67.48	84.16	84.46
	Thermocouple Δ BTU Check	12099.3	11835.4	11828.1	12078.6	12001.7	11819.5
50%	Gas Input BTU's	26476.4	25538.8	25792.4	25831.3	25769.4	25429.9
	Return Air Monitor BTU's	18652.0	18035.7	18385.5	18154.9	18103.4	18004.7
	Room Air Delivery BTU's	20147.3	19016.9	19313.3	19133.6	19043.4	18945.1
	Return Air Efficiency	70.5	70.6	71.3	70.3	70.3	70.8
	Room Air Efficiency	76.1	74.5	74.9	74.1	73.9	74.5
	Stack Flow Rate*-Burner Off	19.43	65.58	63.85	21.32	63.25	66.65
	Stack Flow Rate*-Burner On	63.75	79.83	80.57	63.44	82.21	80.58
	Thermocouple Δ BTU Check	20737.8	20304.5	20874.2	20517.6	20517.6	20454.5
75%	Gas Input BTU's	38128.1	38371.9	39138.2	38709.9	38633.8	38344.4
	Return Air Monitor BTU's	27913.3	28377.8	28193.5	28128.3	29409.7	27295.2
	Room Air Delivery BTU's	29434.6	29761.4	29528.0	29530.6	30893.5	29334.9
	Return Air Efficiency	73.2	73.4	72.0	72.3	76.1	71.2
	Room Air Efficiency	77.2	77.6	75.5	76.3	80.0	76.5
	Stack Flow Rate*-Burner Off	22.35	62.78	65.36	N.A.	59.02	60.80
	Stack Flow Rate*-Burner On	59.35	76.35	78.36	54.30	82.53	77.01
	Thermocouple Δ BTU Check	30360.3	31472.7	32118.1	31646.7	31613.3	31472.6
100%	Gas Input BTU's	50427.9	50519.6	51388.1	50485.4	50817.2	52146.1
	Return Air Monitor BTU's	37142.1	37423.1	37234.7	37435.9	37901.5	36894.3
	Room Air Delivery BTU's	39833.9	39177.9	38949.9	39186.8	39655.8	39166.1
	Return Air Efficiency	73.7	74.1	72.5	74.2	74.6	70.8
	Room Air Efficiency	79.0	77.6	75.8	77.6	78.0	75.1
	Stack Flow Rate*-Burner Off	-------	-------	-------	-------	-------	-------
	Stack Flow Rate*-Burner On	53.48	77.63	77.07	52.75	75.65	77.31
	Thermocouple Δ BTU Check	41338.8	41725.2	42084.4	41996.3	42098.1	42324.2

*Flow rate is in ft^3/min

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